MACHINE DESIGN

November 1944

In This Issue:

Specifying Dynamic Balance

Influence of Bearing Finish

What Does It Take To Build Great Motors?

1 **FOR ONE THING,** it takes a lot of *quality* men like Bill Seebode, 50 year man at A-C. There's no machine known that can assemble the maze of wiring and insulation that goes into a stator with Bill's skill and care ... or that can fully test how *well* he's done his job. There's only one test ... wait 5, 10, 15 years and *see.* And that's the test that has proved over the years, Allis-Chalmers motors are *great* motors!

2 **HUNDREDS OF A-C MEN,** proud that they have won and are *keeping* the confidence of engineers and operating men throughout American industry, have a *personal* stake in every motor they build for you. That's one reason why Stan Brothers follows his job so intently ... as he painstakingly polishes a shaft down to a finish slicker than silk, getting rid of every irregularity that could someday invite failure.

3 **TO WATCH HIM OPERATE,** you'd think Walter Conner was conducting a one-man scrap drive. Not that he finds many defective motor shafts in the course of a day's inspection — but he sure *looks* for them. And when he finds one — bang — it lands on a scrap pile. In a way, we feel it's the group of motors we *didn't* build that's behind the reputation: "You can *depend* on Allis-Chalmers Motors!"

4 **NO—THAT'S NOT A SICK MOTOR.** Actually it's a very *healthy* one. But that's something we must know, through generous sample testing, about every motor before it can tackle a job for you. Part of the "physical exam" McGarvey Eddleman is giving this motor is the heat-run test ... checking speed and temperature hourly while motor runs at full load. It's just one precaution that helps A-C build consistently great motors.

5 **YES, MEN MAKE MOTORS!** Men like Sam Meister, for example. The machine doesn't exist that can duplicate his skill. With an acetylene torch in one hand, a silver alloy rod in the other, Sam silver-brazes the end connections on famous Allis-Chalmers rotors. Round and round the connections he works — expertly flowing in molten alloy to form a joined structure that can withstand as much heat as though it were a single die-casting.

IF YOU COULD MEET and talk with the men who build Allis-Chalmers motors, you might be surprised to learn how keenly they are aware of the big *personal* stake they have in every motor they build for you.

They know that factory tests to fully pre-determine how well a motor is built *just don't exist;* that there's still no substitute for responsible craftsmanship.

And they know that when they build *great* motors for you they're making *friends* — and that friends are something no company and its workers can have too many of.

Next time you need the kind of extra-quality (but not extra-price) motors Allis-Chalmers builds, check with our nearby district office. Or write direct to ALLIS-CHALMERS, MILWAUKEE 1, WIS.

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YOU CAN DEPEND ON ALLIS-CHALMERS MOTORS

MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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NOVEMBER, 1944 Volume 16—Number 11

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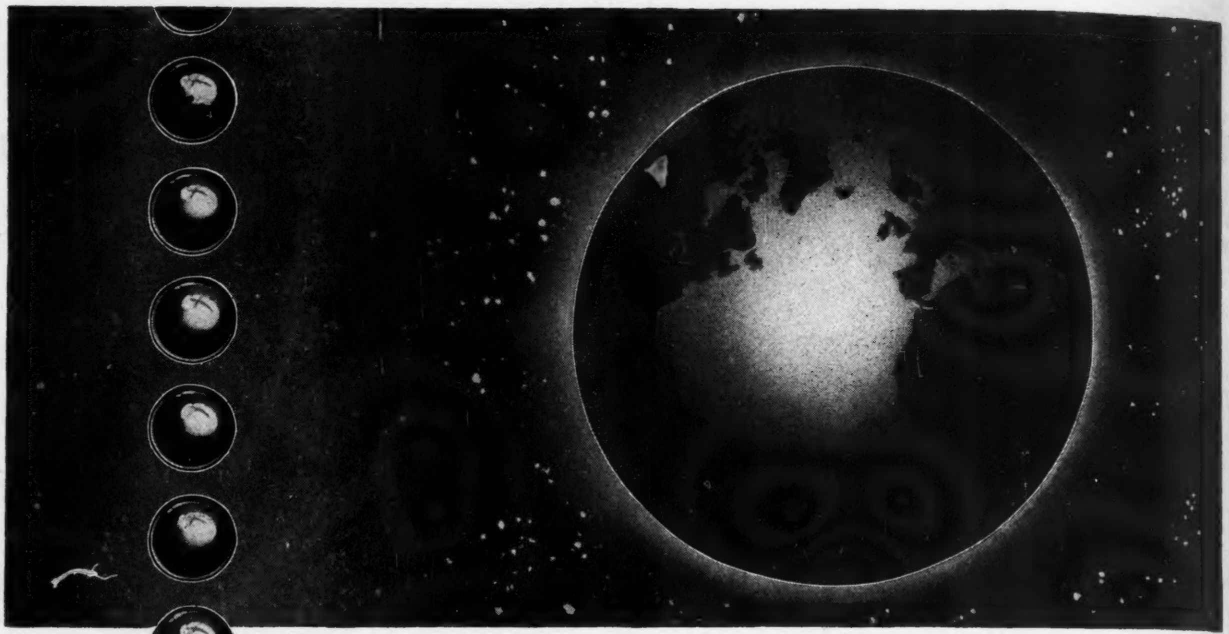

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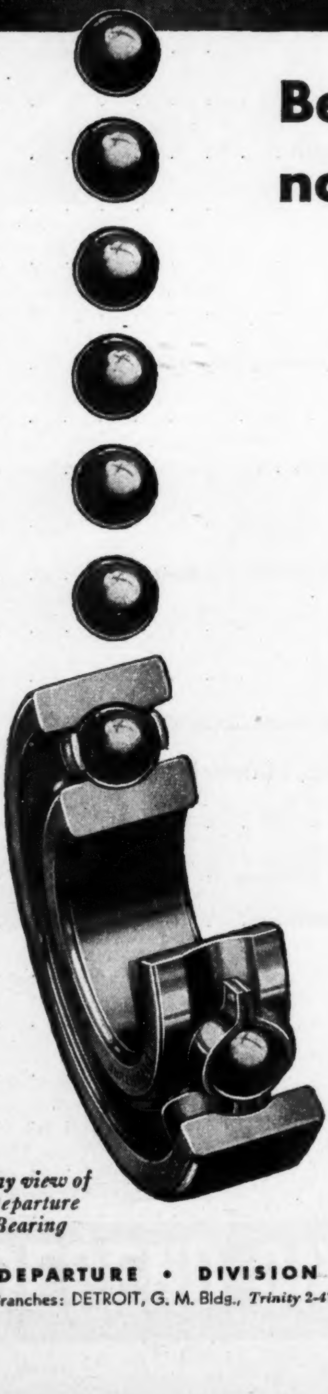

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Itemized Index

Classified for Convenience when Studying Specific Design Problems

MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.

This issue at a glance

Shimmy May Stop the Show in a Theatre . . . .

. . . . but when it stops a production machine it's subject to censure. True, destructive vibration can be eliminated by accurately balanced machine members. But how to specify in clear, simple language the balancing procedures required in the shop? See Page 101 for the first article of a series on this timely subject by W. I. Senger.

Putting More Muscle Into Machine Members . . . .

. . . . is the job of shot peening. This comparatively new shop procedure is bound to exert great influence on design of the future. Just what does shot peening offer and what is the scope of today's knowledge? Don't miss this article. You'll find it on Page 145.

Specifying Steels on "By Guess and by Gosh" Basis . . . .

. . . . won't do for postwar machines. They'll be faster, more compact, more precise and more automatic. Steels used in them will have to be scientifically selected. Get the story on selection based on hardenability from A. L. Boegehold. First of a two-part series commences on Page 129.

Electronics and the Gyroscope Have Combined . . . .

. . . . in the automatic gyro-pilot to help give our fighting airmen supremacy. You'll want the engineering facts behind this outstanding application of electronics. They are clearly and authentically discussed in the article appearing on Page 117.

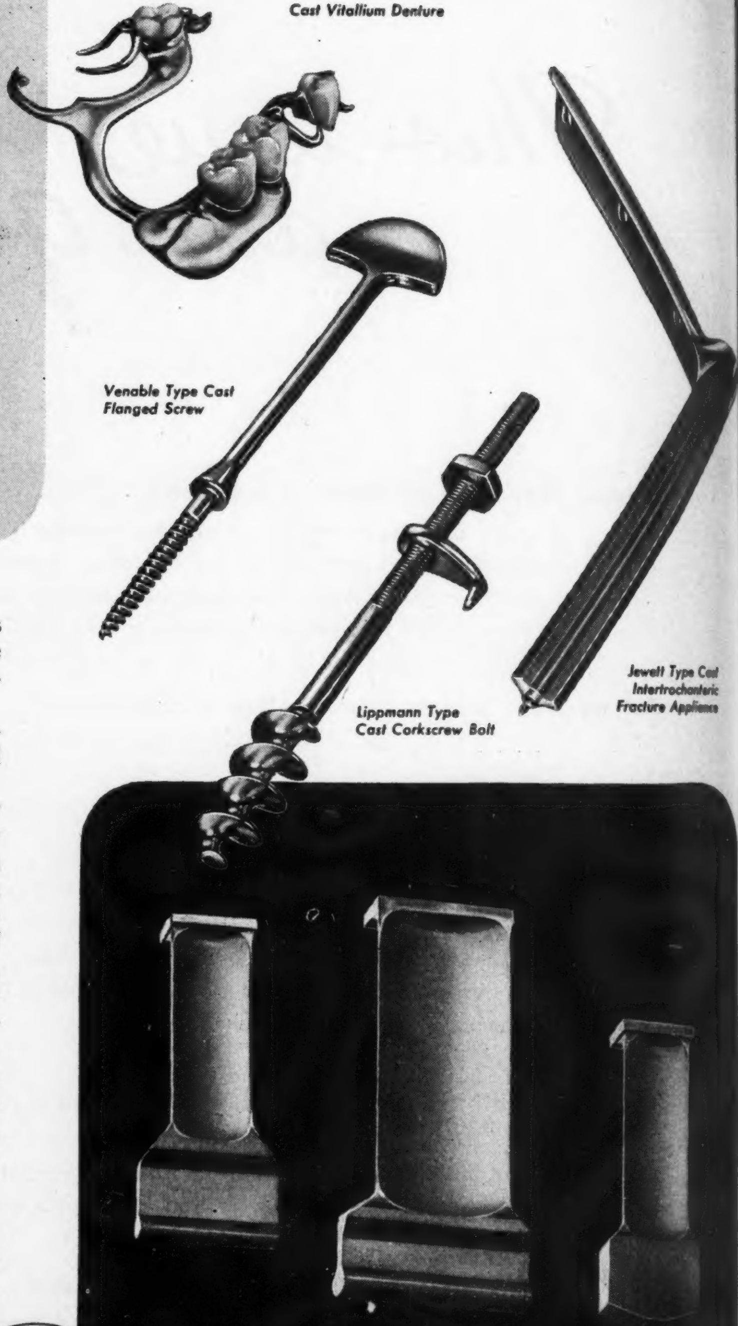
Better Plain Bearing Performance Will Be Needed . . . .

. . . . to meet future demands for increased speed. How can it be improved? Surface finish is a factor of profound importance. Commencing on Page 123, E. L. Hemingway gives you the results of his recent revealing research on the influence of shaft finishes.

"Ease of Operation". That's the Keynote . . . .

. . . . to proper location and design of hand controls. There are other factors too. The subject of hand controls on machines is one of ever-increasing significance. It's influence on worker fatigue and production is tremendous. Turn to Page 111.

From Dental and Surgical Appliances To TURBINE BLADES* For U. S. Warplane SUPERCHARGERS



TURBINE BLADES CAST BY MICROCAST PROCESS (Actual Size)

SIXTEEN YEARS AGO Austenal Laboratories originated the MICROCAST PROCESS for the precision casting of intricate dental appliances to close tolerances.

Limited scale production on surgical appliances and jewelry specialties was started as far back as eight years ago.

For the past three years, the MICROCAST PROCESS has been adapted to the quantity production of many millions of precision blades for supercharger wheels on U. S. high altitude bombers and fighters. *This has resulted in better operating performance and lower production costs on these cast parts, which meet specified dimensions with little or no machining.*

While this new process does not displace or limit conventional forging and machine shop methods, it represents a distinct step forward in casting procedure particularly in the field of non-forgeable and non-machineable metals and alloys.

The MICROCAST PROCESS has already been adopted by a number of leading industrial concerns because it offers new efficiencies, economies and improvements in the mass production of precision castings.

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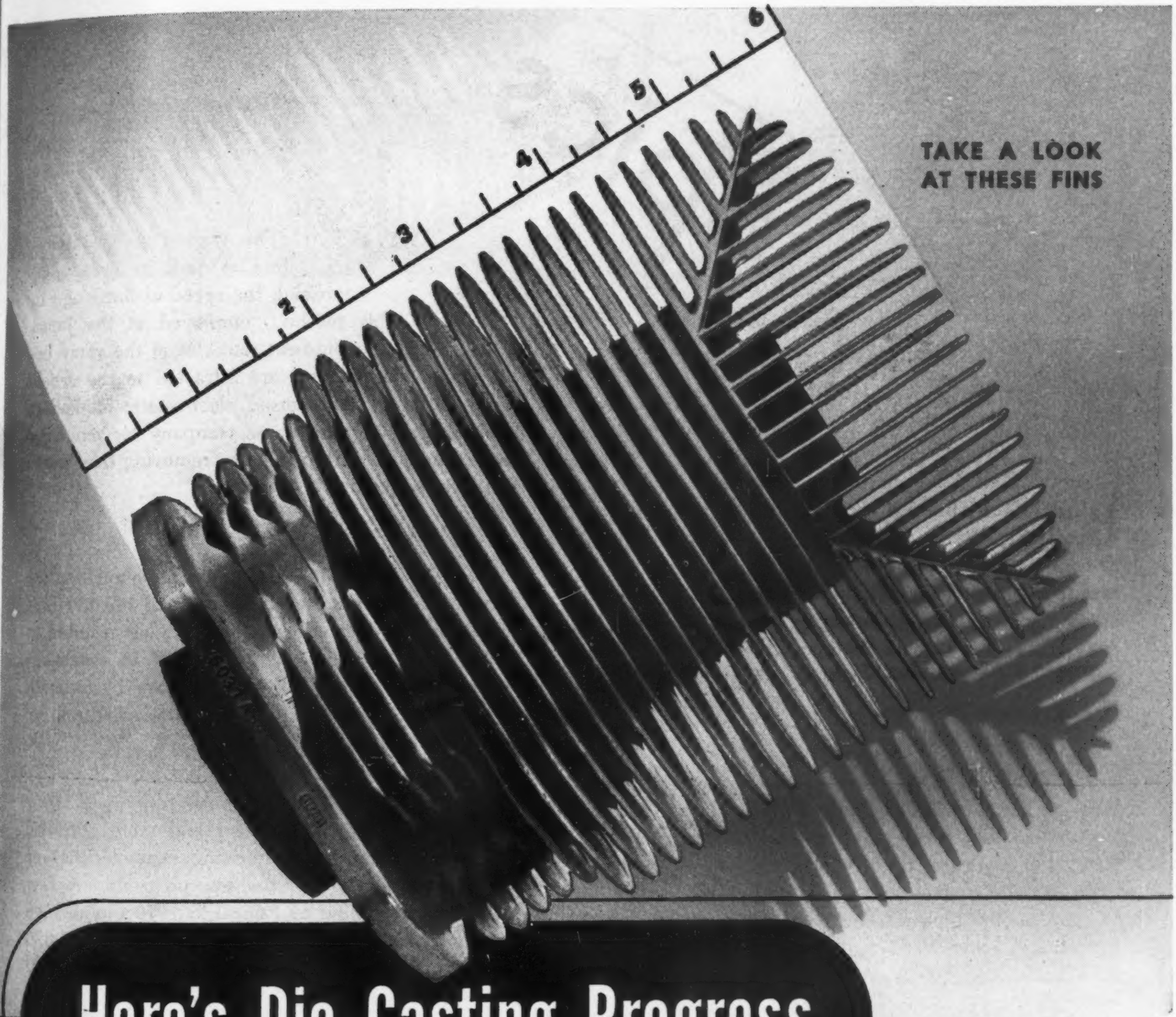


Here's Die Casting Progress

Thinner, deeper and more closely spaced fins have been the trend in die castings like this, designed for rapid heat transfer. Alcoa's Garwood, New Jersey, Plant has pioneered in designing light-metal parts for production by this process, in making dies to produce these increasingly difficult designs, and improving casting techniques.

Manufacturers of engines and compressors have been able to obtain better, more efficient performance from equipment employing these Alcoa die castings. Since all of these products are materials of war, this progress has meant a boost for that effort.

Contributing, too, are these factors: Aluminum provides light weight with high strength, ability to withstand high temperatures, permanence of dimensions and shape, and resistance to corrosion. The die casting procedure speeds up production through accurate rendering of every design detail, with close tolerances, cast-in inserts, thin sections and smooth surfaces. Man-hours are saved, because a minimum amount of machining is required.

Doubtless, there are uses for which you haven't been able to obtain aluminum. Winning the war comes first. But aluminum is now being used for other-than-war purposes, as the manpower situation permits. Our representatives will be glad to discuss the availability of aluminum with you. ALUMINUM COMPANY OF AMERICA, 1940 Gulf Building, Pittsburgh 19, Pa.

ALCOA ALUMINUM

REG. T. M.

Topics

PLASTIC BUBBLES, produced by atmospheric pressure, form crystal clear "teardrop" enclosures from single sheets of Lucite methyl methacrylate resin for P-31 Mustang fighter planes. The enclosures are drawn in a vacuum die to eliminate contact of the plastic with molding dies. The method preserves the optical properties of the plastic.

AERODYNAMIC braking is now being successfully applied to large multiengine land planes. Recent army tests with two and four-engine planes show that reversing the pitch of propellers can reduce materially the landing roll of these planes. Weight savings through the reduced size of wheel brakes and lower loads on landing gear will be reflected in increased payload.

GERMAN arms and equipment captured by American troops in France and the Low Countries will soon be used against the Nazis. As much as fifty per cent of this equipment has been salvaged.

WITHOUT WAR PAINT the Curtiss C-46 Commando is a faster ship and can carry a greater payload than when camouflaged. The reduced weight through the elimination of paint amounts to more than 75 pounds. Also aerodynamic drag presented by the comparatively rough surface of the paint is eliminated, allowing better plane performance.

Salvaging hard rubber from worn tires or treads is effectively performed with induction heating units. The method heats the metal base sufficiently to destroy the bond, enabling the rubber to be stripped off.

SPRAYING of pigmented paints, lacquers or enamels in an electrostatic field conserves as much as fifty per cent of the finish that would otherwise be wasted to the atmosphere as well as greatly increasing the speed of finishing. In this method, employed at the Japan Co., the pigment particles in the spray become charged and are attracted to the article being finished. A reversed electrostatic method is also employed by the same company for removing excess paint from dipped parts, removing the "tears" and fatty edges on the parts.

SIMPLE ADDITION of crankcase ventilation for automobile engines, used in some 28,000,000 cars since the improvement was devised, has resulted in extra value through longer life and in repair-bill savings amounting to billions of dollars, by estimate of H. G. Weaver, director of customer research for General Motors.

DEVELOPMENT of bigger and more powerful planes has been the outstanding characteristics of the nation's aircraft production program. A few months after the fall of France in 1940 the average airframe weight of American warplanes was 3,020 pounds in contrast to the present weight of 10,270 pounds.

JET PROPULSION aircraft turbines will be built in General Electric's second largest wartime plant, constructed two years ago. The entire plant of 600,000 square feet of manufacturing space will not be sufficient to meet the government's orders so another large corporation will also build the jet turbines.

MOST PREVALENT failures in the electrical systems of combat aircraft are caused by vibration of main-engine-driven generators. Based on General Electric Co. tests a stiff mounting flange and a flexible drive assembly having a satisfactory damping device are primary requisites for reducing this vibration.

ENGINEERING STATISTICS and structural metallography are two new courses offered by the graduate school of Stevens Institute of Technology. In addition other engineering courses are being expanded, notably in powder metallurgy, despite the uncertainties of wartime conditions.

MACHINE DESIGN

SPECIFYING DYNAMIC BALANCE

Part I—Types of Unbalance

By W. I. Senger
Balancing Machine Division
Gisholt Machine Co.

DURING the war period the use of static and dynamic balancing machines in production has increased at a tremendous rate. Before the war, balancing machines were available for parts weighing from a few pounds to less than 2000 pounds. Now balancing of parts weighing a fraction of an ounce to 100,000


Fig. 1—Typical rotating parts requiring balancing to eliminate objectionable vibration effects

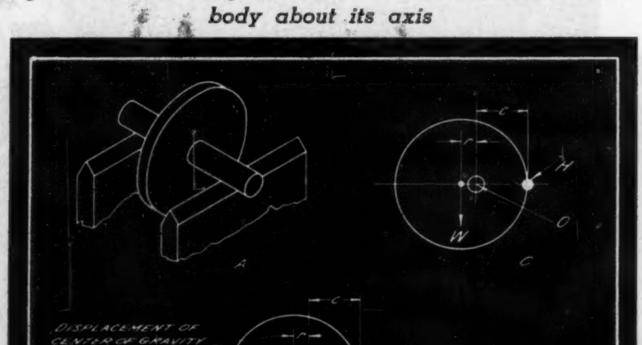
pounds is common and the range is continually being increased.

Unfortunately, general knowledge of the problems involved in the balancing of rotating parts has not increased at a rate commensurate with the increase in use of balancing machines. The failure of engineering groups to understand and give due consideration to the manufacturing problems associated with balancing and the failure of the manufacturing groups to appreciate the engineers' specifications for balance have led to much confusion and misunderstanding. Perhaps the lack of a common language for expressing balancing requirements and tolerances in a form understandable to all parties is responsible for much of the present difficulty. It is hoped that this series of articles will provide a means whereby the balancing problems of either the engineer or the mechanic may be understood by the other party so as to obtain maximum improvement in product with maximum production.

Let us consider in turn (1) the kinds of unbalance which may exist in a rotating body, (2) the causes for the presence of this unbalance and factors influencing or tending to change the condition of balance in a rotating body, (3) rational methods of correcting a condition of unbalance, and (4) equipment available for measuring and locating the correction material which must be added to or removed from a rotating body to give balance. The first factor will be the subject of the present article and the others will be discussed in following articles of this series.

COMMON LANGUAGE between the engineering and production departments is essential in elimination of confusion and errors. In the past, basic understanding has not existed with respect to dynamic balancing. The series of articles of which this is the first presents a concise method for specifying balance on production drawings in such a way as to indicate allowable tolerances without complicating information with technical details

Fig. 2—Schematic representation of static unbalance of a body about its axis



In rotating bodies, unbalance is often the cause of vibration effects which are objectionable to the user of a device, cause of bearing or structural failure, cause of power loss or lowered efficiency, and the like. Of the two general types or forms of unbalance, each causes a different type of vibratory motion of a rotating part. The first type is static or force unbalance and the other dynamic or moment unbalance. Typical parts requiring various degrees of dynamic balance are shown in *Fig.* 1.

Magnitude of Static Unbalance

Static unbalance of fairly large magnitude may be easily demonstrated in a body if the body is supported on its bearing points on two horizontal parallel bars or knife edges as shown in *Fig. 2a*. The heavy or unbalanced part of the piece will always seek a position vertically below the point of contact of the bearings with the parallel bars. The reason for this tendency of one point to seek a lowest position will be evident by reference to *Fig. 2b*, where B represents the point of contact of the bearing with the knife edge and r represents the displacement of

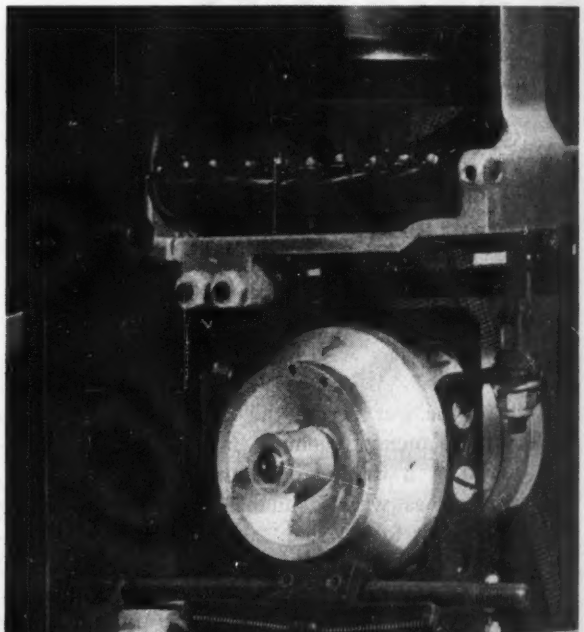


Fig. 3—Delicate instrument parts such as this gyro for a gun stabilizer must be balanced extremely accurately

the center of gravity of the body from the rotational axis. The body when supported at B will have a tendency to turn or roll proportional to the weight of the body W (the pull of gravity on the mass center) multiplied by the displacement r. This tendency to roll may be removed by adding a weight H at a distance c from B such that

$$Wr=Hc$$

Thus static or force unbalance is actually a displacement of the center of gravity of a body from its rotational axis, for if there were no displacement there could be no ten-

dency to roll. The causes of such a displacement will be discussed later.

A more theoretical approach to a condition of static unbalance may be had by reference to *Fig. 2c*. If a body of weight W has its center of gravity displaced a distance r from the rotational axis O, and if the body is rotated at a speed of N revolutions per minute, the centrifugal force developed will be

$$F=\frac{Wr}{g}\left(\frac{2\pi N}{60}\right)^2$$

where g is the gravitational constant. This centrifugal force may be counteracted by adding a weight H opposite in direction from the displacement of the center of gravity and at a distance c from the rotational axis such that

$$\frac{Hc}{g}\left(\frac{2\pi N}{60}\right)^2=F=\frac{Wr}{g}\left(\frac{2\pi N}{60}\right)^2$$

From this equation we obtain the interesting formula $Hc=Wr$. Thus the correction required for static unbalance is a function of the weight of the part multiplied by the displacement of the center of gravity from the rotational axis. If the ounce be used as a measure of weight and displacement be measured in inches, the unit in which static unbalance corrections must be measured is ounce-inches, the product of weight and displacement.

Balancing Tolerance Dependent on Weight

Closer consideration of the above formula will make it evident that the accuracy to which a body may be statically balanced will vary directly with the weight of the body, assuming the same ability to measure displacement of the center of gravity. To say this another way: The heavier the body, the greater must be the balancing tolerance, all other things being equal. Further, from the formula, it should be evident that anything which causes a variation in the displacement of the center of gravity from the rotational axis while a body is rotating must in itself introduce unbalance effects.

When a statically unbalanced body is rotated, our laws of mechanics tell us the body will tend to rotate about its center of gravity. Therefore, if the bearings supporting the body are free in space, the center of the bearings will be caused to describe circles when the body is rotated and at any instant the direction of this displacement will be the same for both bearings. It is this tendency of a statically unbalanced body to produce displacement of its supporting bearings which causes vibrations that are undesirable.

The second type of unbalance which may be present in a rotating body is dynamic or moment unbalance. This is important in small precision parts, *Fig. 3*, as well as in heavy, massive units. The unbalance effect can only be noticed when a body is rotating. This type of unbalance effect may be produced by mounting on a shaft two disks, one on each end of the shaft, each disk having the same physical dimensions and each being mounted eccentrically on the shaft, the direction of eccentricity being opposite on one disk with respect to the other, *Fig. 4*. If such a

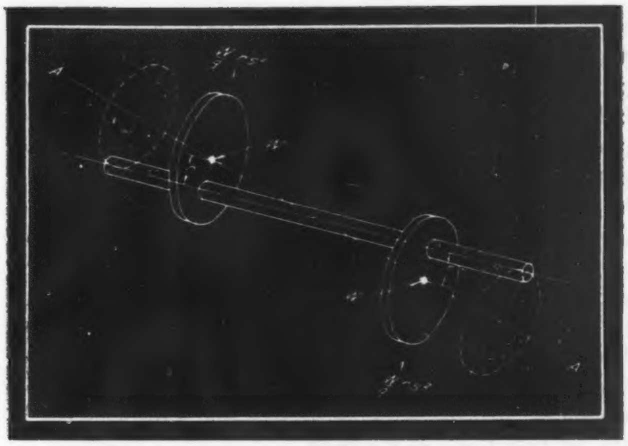
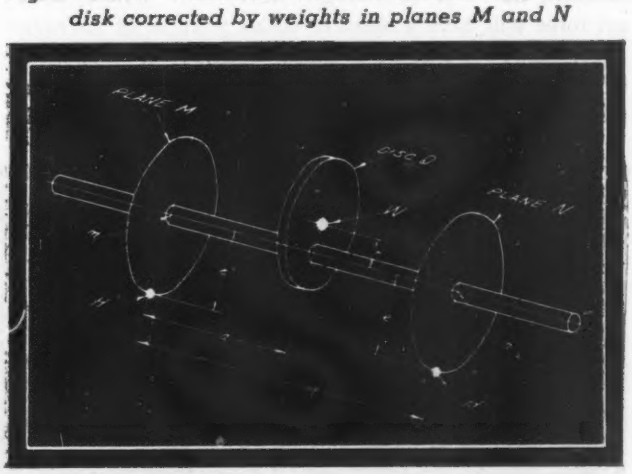


Fig. 4—Dynamic unbalance of two disks would, if shaft is unrestrained, cause rotation about A-A

Fig. 5—Below—Unbalance effect of eccentrically mounted disk corrected by weights in planes M and N



body be rotated, the rotation will be about an axis A-A if the shaft is not restrained in bearings (assuming that the shaft has no mass). Axis A-A is a principal inertia axis of the body. The rotational axis of the shaft will actually generate two cones, one end of the shaft always being displaced on the opposite side of axis A-A from the displacement of the other end of the shaft. If the shaft is restrained, the couple of forces tending to displace the shaft will cause vibration. (The forces are indicated by the arrows marked $W\,r\,s^2/g$.)

Series of Disks Used as Basis

Any rotating body may be considered as made up of a multitude of disks, and the center of gravity of each of these disks may or may not be eccentric with respect to the rotational axis as determined by the bearings supporting the body. The displacement of the supporting bearings of such a body will be determined by the resultant effect of each of the eccentric disks on the entire body. As it is reasonable to suppose that each disk will be eccentric and that the amount and direction of the eccentricity will vary, it is also reasonable to suppose that any rotating body not a perfect disk will be both statically and dynamically out of balance.

As has already been stated, any body may be consid-

ered as made up of a multitude of disks, each of which may introduce an unbalance effect. If it is possible to prove that any one of these disks which is producing unbalance in a body may have its unbalance effect compensated for by the addition of weights in two transverse planes, it should be obvious that a similar procedure may be followed to determine compensation for every other disk in the body, and that the resultant or summation of all of these effects in each of two transverse planes will actually be a measure of the amount of correction needed to eliminate both static and dynamic unbalance from the body. In *Fig.* 5, disk D is intended to represent one of many eccentrically mounted disks on the shaft. It has a weight W, and the center of gravity is at a distance r from the rotational axis of the shaft.

To correct for the unbalance effect Wr, weights may be added in the planes M and N which are a distance l apart, the disk D being a distance a away from the correction plane M. The centrifugal force produced by the eccentric disk will be $(Wr/g)\ (2\pi N/60)^2$. The centrifugal force will have a moment about point m of (Wra/g) $(2\pi N/60)^2$. We desire to find a weight H at a distance e from the rotational axis in the correction plane N having the same moment about point m. The moment of the force produced by H will be $(Hel/g)\ (2\pi N/60)^2$. Equating these two moments,

$$\frac{Wra}{g}\left(\frac{2\pi N}{60}\right)^2=\frac{Hel}{g}\left(\frac{2\pi N}{60}\right)^2$$

and simplifying this equation,

$$Wra=Hel$$

This may be otherwise stated: The unbalance correction in Plane N in ounce-inches equals $He=Wra/l$. In a similar manner, the correction required in plane N in ounce-inches equals $H'e'=Wr(l-a)/l$.

Corrections Compensate for Unbalance

It can be proved algebraically that the sum of the forces added by the introduction of a correction H and H' is equal to the force produced by the eccentrically mounted disk W and that the moments of these forces about any point is zero. Thus the corrections H and H' will permit rotation of the body in its bearings without introducing any centrifugal forces or moments tending to displace the rotational axis from the axis determined by the supporting bearings. It is this displacement of the rotational axis, due to the unbalance, which produces the vibration effects previously mentioned.

Therefore, before proceeding further, a formula should be developed for the displacement of the rotational axis of a balanced rotating body due to the introduction of an unbalanced weight. Because bearing displacement is to be eliminated by balancing operations, such a formula should provide a means for determining the physical properties of a body, such as shown in *Fig.* 6, that will determine the bearing displacement. When this has been done we will have a means for determining the accuracy required to reduce bearing vibration to a desired value.

The body shown in *Fig.* 7 represents any body in which it is desirable to determine the displacement of the rotational axis from the principal inertia axis if an unbalance

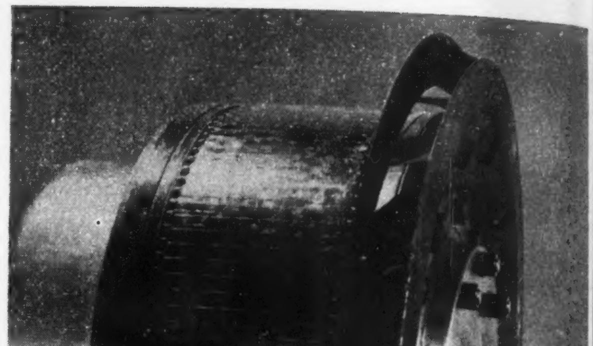


Fig. 6—Above—Motor armatures from minute one-ounce instruments to the 6000-pound armature shown must be dynamically balanced to improve operation and increase life. This armature is balanced within ½-ounce at ring on inside of commutator

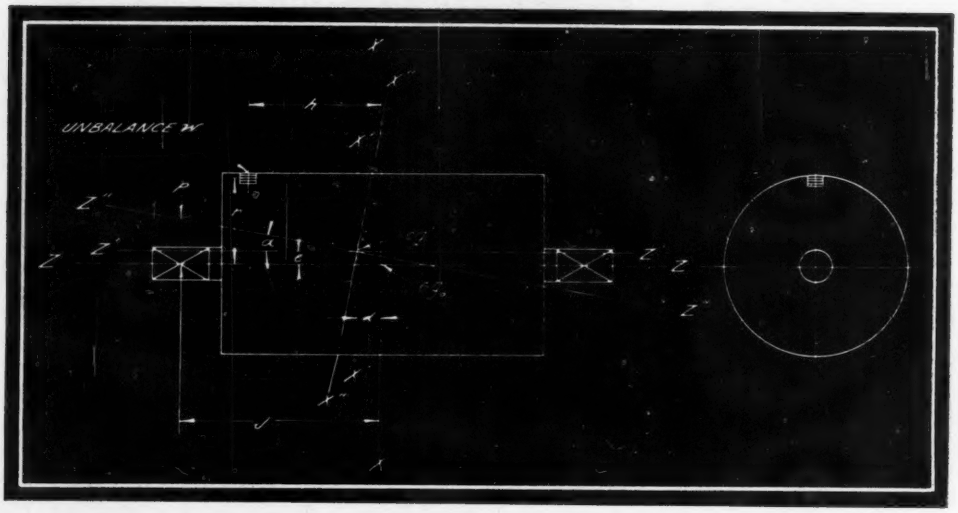


Fig. 7—Left — Diagrammatic representation of displacement of rotational axis resulting from unbalance

is added at a certain point. Assuming the weight of the body is W and the principal inertia axes are X-X and Z-Z as shown, an unbalanced weight w (w to be small as compared to W) is added at a radius r from the axis Z-Z and at a distance h from axis X-X. The addition of this weight will cause a translation and a rotation of the principal axes of inertia so that the new principal axes will be X''-X'' and Z''-Z''. Under rotation this part, if not restrained, will rotate about the principal inertia axis Z''-Z''. Therefore, we are interested in determining the displacement of axis Z''-Z'' from the rotational axis Z-Z, for it is this displacement which produces the unwanted vibrations when the part is restrained in bearings.

If I_z represented the moment of inertia in mass units about axis Z-Z before the weight was added, the moment of inertia after adding the weight will be

$$I_{z_w} = I_z + \frac{wr^2}{g} \qquad \text{(A)}$$

Similarly, if I_x represented the moment of inertia about axis X-X before the weight was added, the moment of inertia after adding the weight will be

$$I_{x_w} = I_x + \frac{wh^2}{g} \qquad \text{(B)}$$

Further, if X-X and Z-Z were principal axes before the weight was added, the product of inertia about these axes was zero; but, after adding the weight, the product of inertia will be

$$I_{x_w z_w} = \frac{wrh}{g} \qquad (1)$$

Inertias about the translated axes X'-X' *and* Z'-Z' may be determined also, for, from the translation of axes theorem,

$$I_{x'_w} = I_{x_w} - d^2\frac{W+w}{g} \text{ and } I_{z'_w} = I_{z_w} - e^2\frac{W+w}{g} \qquad (2)$$

The product of inertia about the translated axes is

$$I_{x'_w z'_w} = \int x'z'dM$$

but $x' = x - e$ *and* $z' = z - d$, so

$$I_{x'_w z'_w} = \int (x-e)(z-d)dM$$

$$= \int xzdM - d\int xdM - e\int zdM + ed\int dM$$

$$= I_{x_w z_w} - \frac{dwr}{g} - \frac{ewh}{g} + ed\frac{(W+w)}{g} \qquad (3)$$

Before w was added the center of gravity was at cg_o. After adding w the center of gravity is at cg'. Taking moments about the X-X axis,

$$wh = (W+w)d \text{ or}$$

$$d = \frac{wh}{W+w} \qquad (4)$$

and taking moments about the Z-Z axis,

$$wr = (W+w)e \text{ or}$$

$$e = \frac{wr}{W+w} \qquad (5)$$

This equation is identical with the one previously given for a body which is not in static balance. Substituting these known values of d and e in the equation for the product of inertia about the translated axes (Substituting Equations 4 and 5 in 3),

$$I_{x'_w z'_w} = I_{x_w z_w} - \frac{w^2rh}{g(W+w)} - \frac{w^2rh}{g(W+w)} + \frac{w^2rh}{(W+w)^2}\,\frac{(W+w)}{g}$$

Substituting Equation 1 in above,

$$I_{x'_w z'_w} = I_{x_w z_w} - \frac{w^2rh}{g(W+w)} = \frac{wrh}{g} - \frac{w^2rh}{g(W+w)} \qquad (6)$$

Substituting Equations 4 and 5 in 2,

$$I_{x'_w} = I_{x_w} - \frac{w^2h^2}{g(W+w)} \text{ and } I_{z'_w} = I_{z_w} - \frac{w^2r^2}{g(W+w)} \qquad (7)$$

If the axes be rotated through an angle α, the product of inertia about the new axes X''-X'' and Z''-Z'' will be

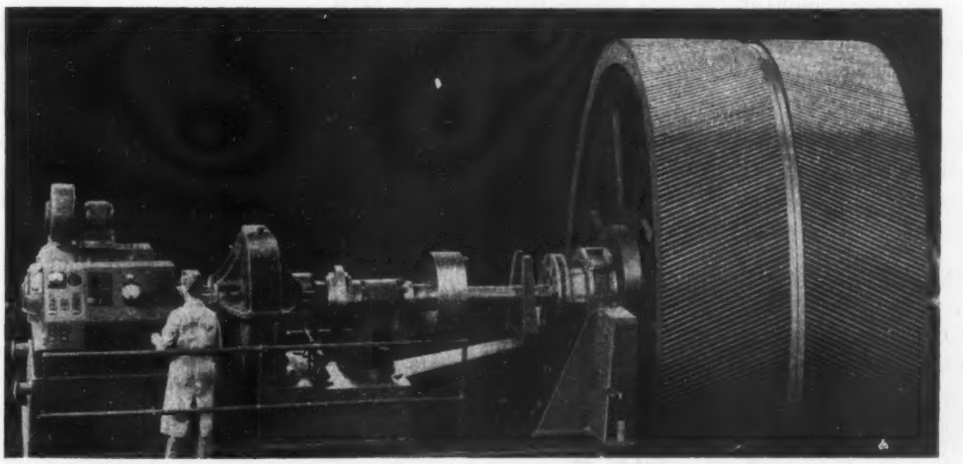


Fig. 8 — Right — Setup for balancing 100,000 - pound ship reduction gears which are balanced within an accuracy of 4-ounce weights on inside of gear rim

$$I_{X''_w Z''_w} = \int x''z''dM$$

$$= \int [(x' \cos \alpha - z' \sin \alpha)(x' \sin \alpha + z' \cos \alpha)]dM$$

$$= \frac{1}{2} I_{Z'_w} \sin 2\alpha - \frac{1}{2} I_{X'_w} \sin 2\alpha + I_{X'_w Z'_w} \cos 2\alpha \quad (8)$$

As the X''-X'' and Z''-Z'' axes are the principal axes after the addition of w, the product of inertia must equal 0 and, therefore,

$$\frac{1}{2} I_{Z'_w} \sin 2\alpha - \frac{1}{2} I_{X'_w} \sin 2\alpha + I_{X'_w Z'_w} \cos 2\alpha = 0$$

$$\frac{\sin 2\alpha}{\cos 2\alpha} = \frac{2I_{X'_w Z'_w}}{(I_{X'_w} - I_{Z'_w})} = \tan 2\alpha$$

Substituting from Equations 6 and 7,

$$\tan 2\alpha = \frac{2rh\left[\frac{w}{g} - \frac{w^2}{g(W+w)}\right]}{I_{X_w} - \frac{w^2h^2}{g(W+w)} - I_{Z_w} + \frac{w^2r^2}{g(W+w)}}$$

Substituting from Equations *A* and *B*,

$$\alpha = \frac{1}{2} \tan^{-1} \times \frac{2rh\left(\frac{w}{g} - \frac{w^2}{g(W+w)}\right)}{I_{X_w} - I_{Z_w} + \frac{w^2(r^2-h^2)}{g(W+w)}}$$

$$= \frac{1}{2} \tan^{-1} \times \frac{2rh\left[\frac{w}{g} - \frac{w^2}{g(W+w)}\right]}{I_X - I_Z + \left[\frac{w}{g} - \frac{w^2}{g(W+w)}\right](h^2-r^2)}$$

If w is to represent a weight equal to the balancing tolerance which will produce negligible vibration, w will be small compared with W. Then, neglecting the terms

$$\frac{w^2}{g(W+w)} \text{ and } \left[\frac{w}{g} - \frac{w^2}{g(W+w)}\right](h^2-r^2)$$

in the above formula introduces negligible errors in most cases. In a few isolated cases such as a sphere, or any rotating shape with the inertia values I_x and I_z substantially equal, this approximation cannot be made. Also since α is dependent upon the relative values of w and W it is a small angle and we can assume that $\tan \alpha = \alpha$. Then

$$\alpha = \frac{wrh}{g(I_X - I_Z)} \quad (9)$$

Actual bearing displacement of the bearing nearest the unbalancing weight will be, from *Fig.* 7,

$$p = e + \alpha J \text{ inches}$$

Substituting from Equations 5 and 9

$$p = \frac{wr}{(W+w)} + \frac{wrhJ}{g(I_X - I_Z)}$$

$$p = \frac{wr}{W} + \frac{wrhJ}{g(I_X - I_Z)} \text{ Approximately} \quad (10)$$

From previous considerations, the first term of this equation is recognizable as the translation of the inertia axis due to the presence of the static unbalance wr. The remaining term is the moment effect of the unbalance weight due to the fact that the weight was placed a distance h from the inertia axis X-X (an axis perpendicular to the rotational axis through the center of gravity of the body). Therefore, the moment or dynamic unbalance in a body may be represented by the product wrh. As w is generally given in ounces while r and h are inches, dynamic unbalance must be expressed in ounce-inch-inches. Further, it should be evident from the formula that the greater the distance from the center of gravity of a body to the bearing supporting the body, the greater will be the displacement of the bearing due to a given unbalance. Also, the greater the difference between the moment of inertia about the rotational axis and the moment of inertia about a diameter through the center of gravity, the less will be the bearing displacement.

Methods for Specifying Accuracy

From the discussion so far given, there are several ways in which the specifications for accuracy of balance may be given. The first method would be to state:

> "This part must be balanced statically to an accuracy of .... ounce-inches and must be dynamically balanced to an accuracy of ...... ounce-inch-inches."

A second method of specifying accuracy of balance is based on the fact illustrated in *Fig.* 5 that all unbalance in a rigid rotating body may be compensated for by the addition of correcting weights in each of two selected transverse planes. From Equation 10,

$$wr = \frac{pWg(I_X - I_Z)}{g(I_X - I_Z) + WhJ} \quad (11)$$

From the formula it is evident that a balancing tolerance may be expressed in terms of ounce-inches in correction planes which are at a distance h from the center of gravity. Such a specification on a drawing should indicate the actual planes of correction and give the accuracy to be maintained in each of the indicated planes. This specification for balancing tolerance is more easily understood by the men actually concerned with the balancing operation and is the most desirable of the specifications so far given. Later, a modification of this specification will be given which will be still more easily understood.

In establishing a tolerance such as suggested, due consideration must be given to the physical properties of the body to be balanced; for example, the mass of the reduction gear in *Fig.* 8 compared to delicate supercharger rotors. In other words, the balancing tolerance must be consistent with the weight of the body, the moments of inertia of the body, the distance between the center of gravity and the established plane of correction, the distance between the supporting bearings, and the bearing displacement. Balancing machines have been developed that will indicate bearing displacements of the order of .00000025, this being an accuracy well beyond the accuracy to which shafts are generally ground for roundness and the like.

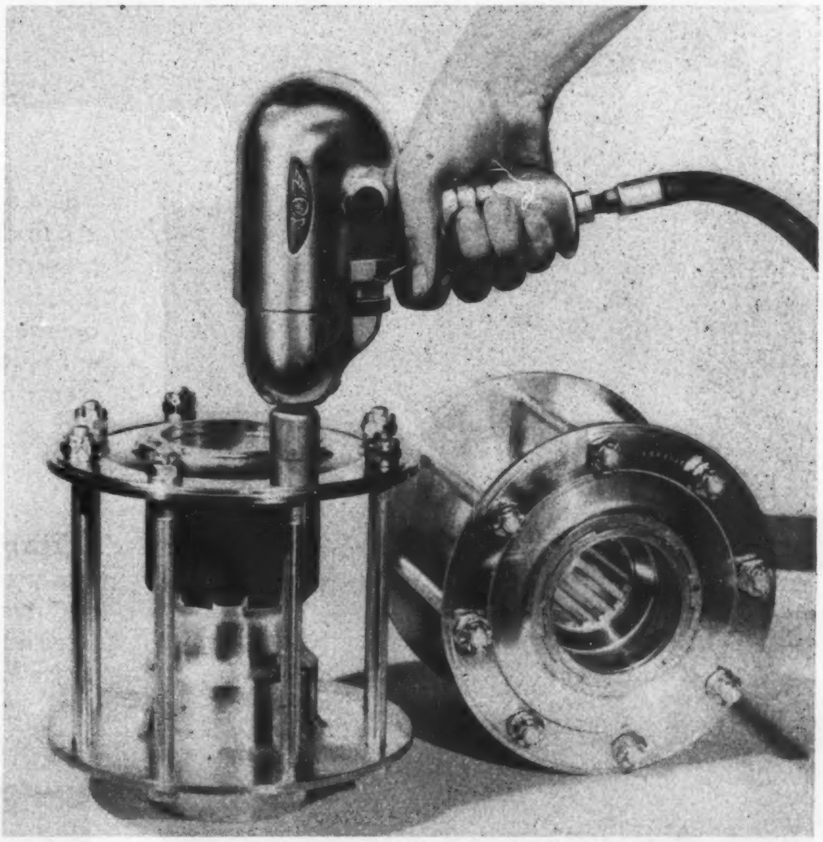
(*Continued in next issue*)

Scanning the Field for Ideas

Torque control is effected in the impact wrench, at right, through the simple hammer and anvil device illustrated in the cross section below. Designed by Aro Equipment Corp., the wrench is driven by an air turbine at speeds between 4000 and 5000 revolutions per minute. Torque delivered is controlled by adjustment of a throttling valve.

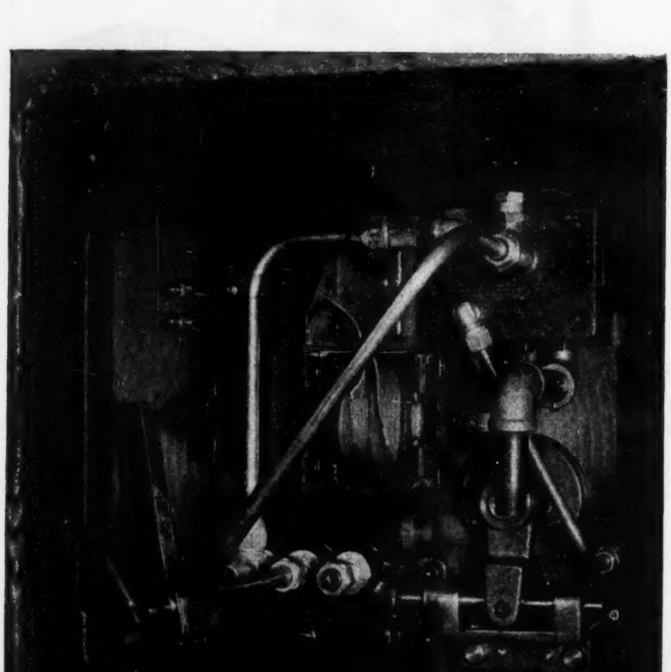
Impact mechanism employs two steel rollers contained in a slot in the rotor in such a way that centrifugal force throws them against the anvil. When the wrench is in operation the

rollers move out to a position where they are fully engaged between the hammer and anvil members. This transfers the full torque of the motor through to the work. When the selected torque is reached, the rollers rebound from the anvil faces. They actually hit and slip by, delivering no additional torque.



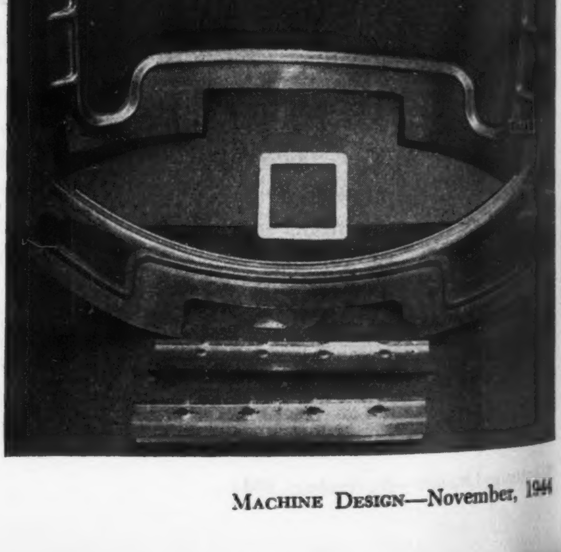
Oil-immersed solenoids for machine control, particularly hydraulic, prove to have several advantages. Oil immersion provides clean operation of solenoids, makes an excellent electrical insulator, dissipates heat from coils, lubricates moving parts, eliminates hammer blow of solenoid core by dashpot effect, obviates packing glands and stuffing boxes, and simplifies operating linkage.

Generally accepted idea that oil is destructive to motors and control is not necessarily correct. Although oil does destroy some kinds of insulation and may be contaminated by cast-iron dust, clean oil in itself is an excellent insulator. Special oils may be employed, but for voltages utilized in machines any good type such as for hydraulic-feed systems is suitable. The National Automatic Tool Co. uses oil-immersed solenoids and has not had a single coil failure due to the oil. A two-way machine and a close-up of one of the reservoirs



where even a film of oil is present, some of the oil is carbonized. On small devices, this sludge will prevent contacts from interrupting the current. If the sludge is washed away the switch will again operate satisfactorily. Switches where oil is present require large opening between contacts and plenty of surface insulation.

Through elimination of eight parts, below, that previously were assembled by spot welding and riveting methods, Lockheed Aircraft Corp. has effected a saving of 112 man hours per airplane in the manufacture of one-piece cowling. Total



with the cover removed are shown in the illustrations above. The right and left beds of the machine have oil reservoirs and all of the equipment shown in the close-up operates completely immersed. Two solenoids are utilized—one for rapid-forward and one for rapid-reverse operation of valves. Terminal connections are modified spark plugs which solved the problem of getting insulated control wires from the conduit system into the oil reservoir.

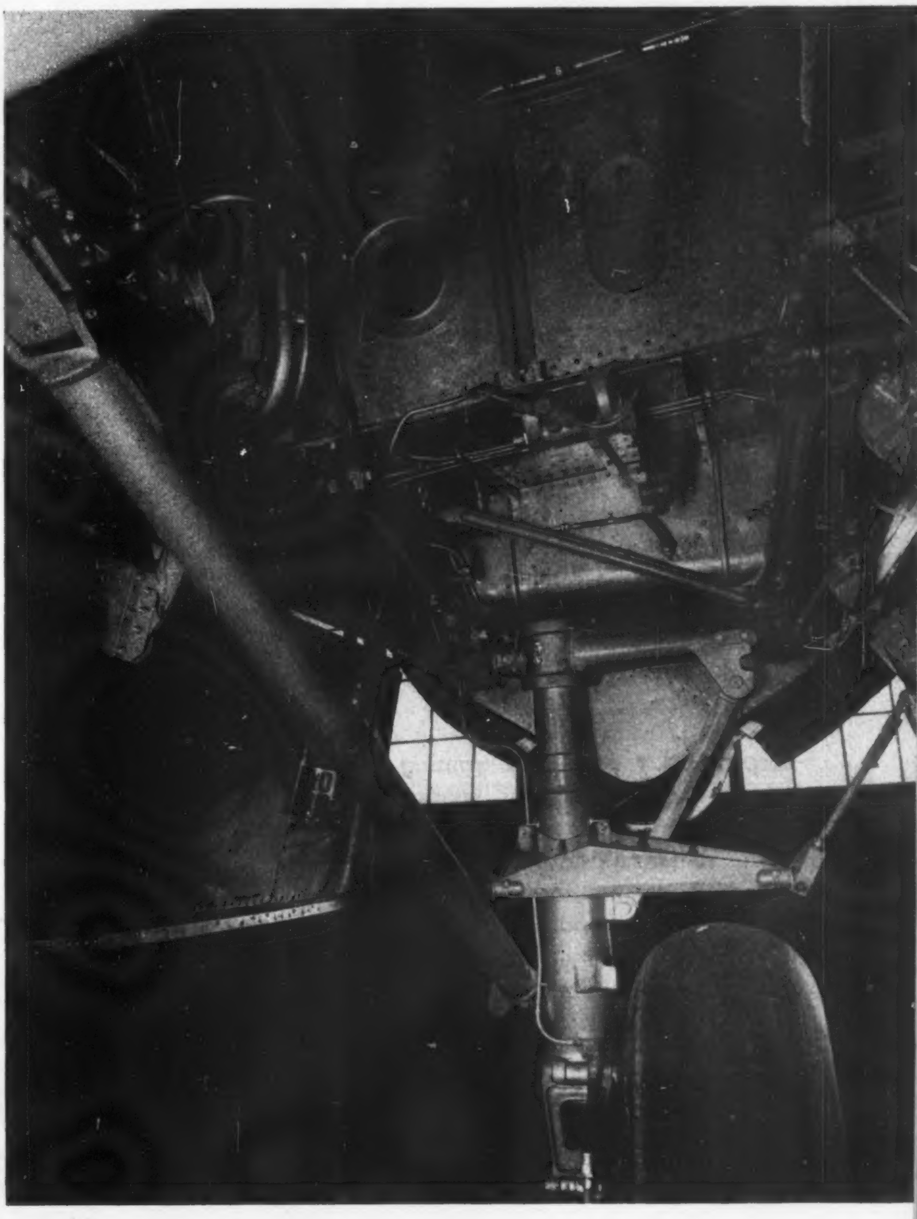
Care in applying other electrical equipment where it is exposed to oil is of course necessary. Application of devices that interrupt current such as relays, contactors, switches, etc., where oil is present introduces another type of problem. Each time current is interrupted,

savings are estimated at $202 per plane and paper work has been reduced so greatly that 16 routing cards are now used where 116 were required previously for a similar cowling fabrication job. The parts shown in the lower portion of the picture are replaced by the new single part above them. The metal cut from the center of the sheet is utilized for fabricating other parts. Lockheed feels that their work simplification program, of which this improvement is a part, has only scratched the surface of possible economies and that substantial savings could be effected in the manufacture of every assembly if each were to be analyzed in a like manner.

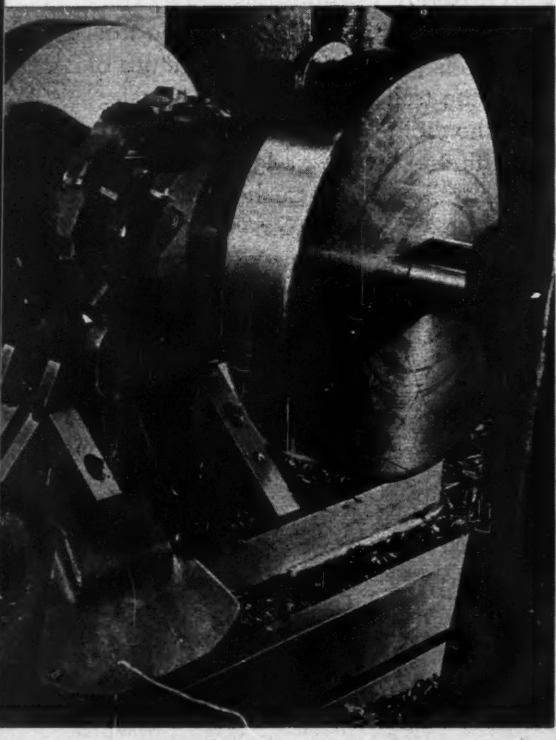
Lightweight retractable landing gear illustrated at right incorporates novel features which permit greater payload and assure proper operation. Proved in service on the A-30 Baltimore, Martin attack bomber, the gear allows a load factor of 6**g** yet is only 4.72 per cent of the plane's weight. The new Martin gear is supported by an N-strut attached to the forward wing spar where it passes through the engine nacelle. A single forging, the strut eliminates many of the small assemblies used in other designs. The oleo is attached to one end of a trunnion shaft turning on bearings in the N-strut and braced with a single cross brace from the other end of the trunnion shaft. Direct connections from the oleo operate the wheel-well doors, eliminating the need for additional hydraulic mechanisms and assuring positive action of the doors.

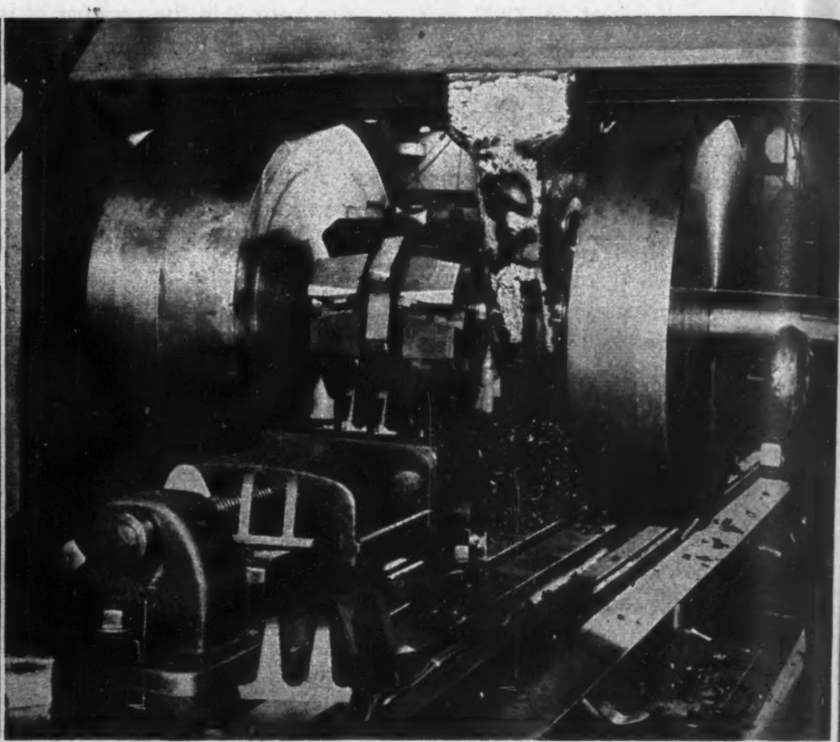
Raising and lowering of the gear is actuated by a hydraulic cylinder that moves an arm attached to the oleo, rotating it about the trunnion shaft as a pivot. An unusual design of the drag strut locks the gear in down position. The forward end of this strut is attached to the oleo while the aft end is fixed to a trolley running on a track on the underside of the wing inside the nacelle. While the gear is in the process of being raised or lowered this trolley runs free but, as the gear reaches the down position, a block on the trolley engages a hook-like fixture which is part of the main structure of the airplane and lifts the wheels of the trolley off the track, transferring the landing or take-off load to the main structure.

At the same time a square pin with one beveled edge, resembling an ordinary door latch, engages the trolley and is pushed into place by a hydraulic cylinder, locking the trolley to the main structure and the gear into down position. When the gear is raised the cylinder withdraws the pin and the trolley slides back along the track as the gear retracts.



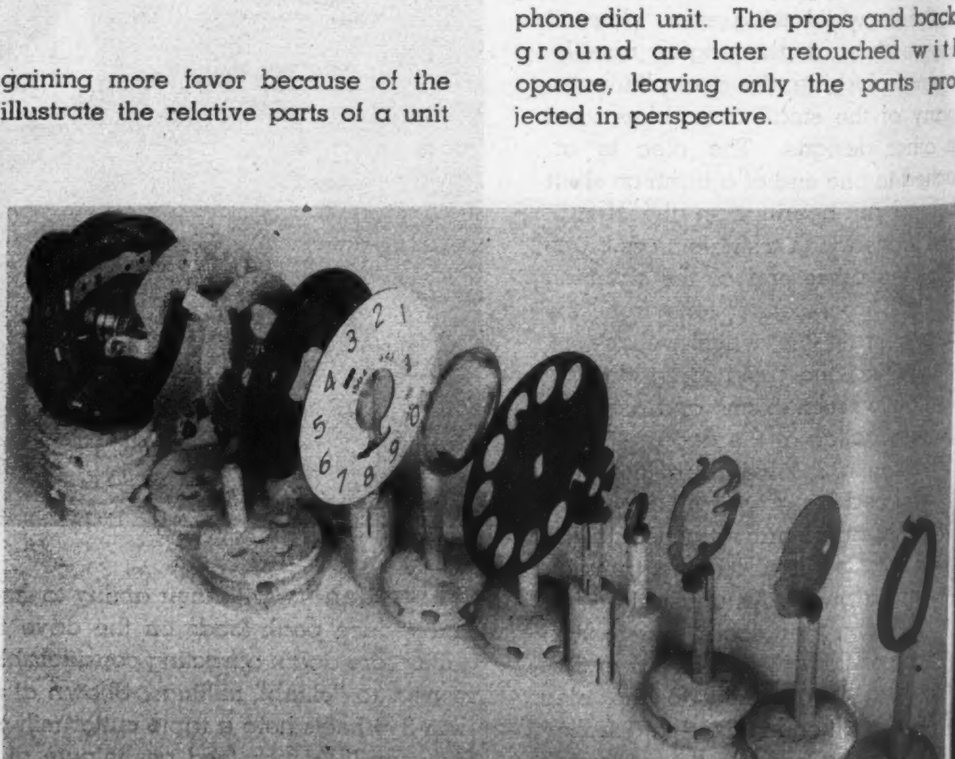
Flywheels, with their ability to smooth out machine operations and reduce peak loads on the drive through their energy-storing ability, are again attracting considerable attention, particularly with respect to "climb" milling. Shown at top of next page at the left, twin flywheels help a triple cutter mill a steel forging at 581 surface feet and $17\frac{1}{2}$ inch feed per minute at the Lockheed plant. In the other photograph, taken at Boeing Aircraft plant, an SAE 4130





steel forging is being milled at 549 surface feet per minute with 10-inch feed with the aid of flywheels to assist the coarse-tooth cutters perform their job smoothly. As shown in these illustrations, the flywheels are placed as close to the source of pulsations as possible. If shafting of any length is interposed the torsional whip of the shaft would tend to reduce the effectiveness of the method.

Exploded views are gaining more favor because of the readiness with which they illustrate the relative parts of a unit and the order of assembly. This pictorial method is especially helpful under current conditions when untrained personnel is being trained to assemble intricate parts. Preparation of exploded views by photographic methods, however, may be tedious because each part must be placed in its proper expanded three-dimensional position. In the illustration below, prepared by Bell Telephone Laboratories Inc., "Tinkertoy" parts provide adjustable props for a telephone dial unit. The props and background are later retouched with opaque, leaving only the parts projected in perspective.



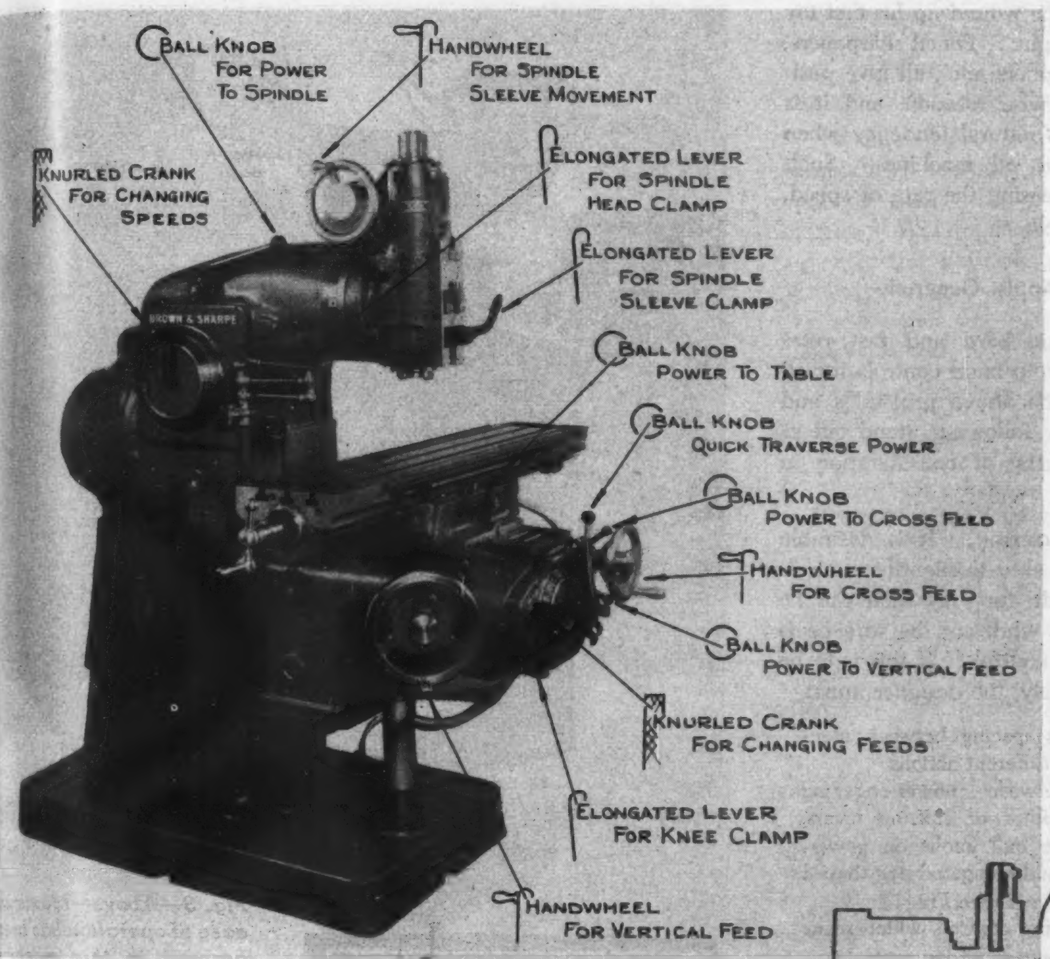


Fig. 1 — Left — Demonstrating how distinction can be made between power-engaging, clamping and shifting controls. Operator facility is thus greatly enhanced

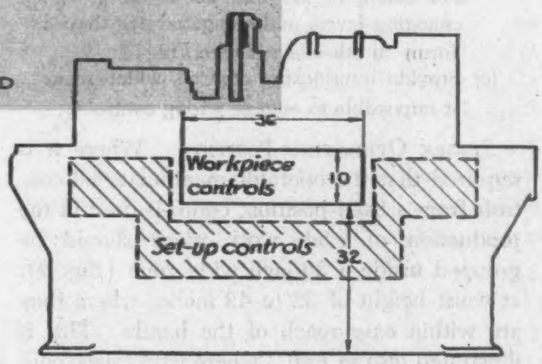


Fig. 2 — Below — Typical arrangement of controls on a machine which is handled by the operator from a fixed position

Are Your Controls Properly Designed, Effectively Placed?

By Harold Sizer

Brown & Sharpe Mfg. Co.

MACHINES of the future undoubtedly will emphasize more than have those of the past such valuable attributes as ease of operation, minimum operator body and eye fatigue, more complete automaticity, and—as a result—better and more efficient performance. The degree to which any given machine will possess these desirable qualities can depend greatly on the care and thoroughness exercised in designing and locating its hand controls.

This subject of the design and placement of hand controls on machines is an intriguing one because it entails not only a knowledge of mechanical design but actually some fundamental understanding of body movements and reactions. For example, most people have better control of and greater strength in their right arm. Thus the discerning designer will place the most sensitive or delicate controls in positions where they can be operated readily by the right hand. Again, everyone has been

taught from the time he wound up his first toy train to "turn to the right". Pencil sharpeners, can openers, meat grinders, etc., all give positive actions for clockwise rotation, and it is wise to recognize this natural tendency when designing controls for all machines. Such actions might be increasing the rate of speed, closing workholders, etc.

Basic Rules Apply Generally

While there are no hard and fast rules which apply inflexibly to hand controls for all types of machines, the above principles and those discussed in the following, stand out as being basic and worthy of consideration in general applications.

Control Identification: It is desirable that the operator be able to identify controls by position and touch, for only then can he keep his eyes on the work and be sure he is operating the proper controls. If this is to be accomplished effectively, the designer must:

(a) Provide generous spacing between identical controls for different actions
(b) Differentiate between power-engaging members and clamps or shifting levers. For example, use ball knobs on power-engaging levers and elongated egg-shaped forms on other levers (see *Fig.* 1)
(c) Provide interlocking controls which make it impossible to engage wrong controls.

Single Operating Position: Where it is required that the operator manipulate all controls from a fixed position, controls used in the production of each work piece should be grouped inside a 36-inch wide zone (*Fig.* 2), at waist height of 32 to 42 inches where they are within easy reach of the hands. This is illustrated also in *Fig.* 3 where work-piece controls are within the prescribed 36-inch zone between the table and crossfeed handwheels while setup controls for table speed, amount of cross feed and truing arrangements, are on the left and right of this zone. It is interesting to note also that the base of the grinder in *Fig.* 3 is recessed to make operation of the machine from a sitting position convenient.

Fig. 3—Above—Maximum ease of operation has been assured in this machine by locating all work-piece controls within a 36-inch zone at waist height. Knee hole permits convenient operation of the machine from a sitting position

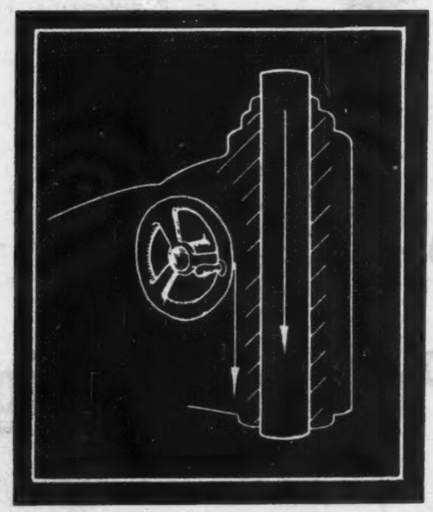


Fig. 4—Left—Portion of handwheel rim closest to eye should move in same direction as the machine motion it inaugurates

Simplicity: The less an operator has to think about hand controls, the more speedily and efficiently will he operate the machine. If several machine actions must start and stop together, they should be operated by a single control. This procedure also results in limiting operator fatigue.

Directional Control: If a handwheel has its axis perpendicular to the direction of the slide movement it produces, the part of the handwheel rim nearest to the eye should have the same direction of movement as the slide. This is demonstrated in *Fig.* 4. In the case of switch buttons, those inaugurating motion to the right should be on the right side of a control post or panel—those for movement to the left, on the left side (see *Fig.* 5). Where levers are employed, the direction in which a lever is thrown should be the direction of the slide movement it controls (*Fig.* 6).

Safety: Design and locate controls so that accidental power engagements or undesired machine actions will not be caused by brushing or leaning against the machine. Where pushbuttons are used this can be accomplished with recessed buttons as shown in *Fig.* 5. The movement required of the operator to stop the machine should be a simple one entailing no exact positioning of hand or finger. A sweep of the arm or blow of the hand (see mushroom-type button of *Fig.* 5) should be all that is necessary.

Keep controls well away from power members such as

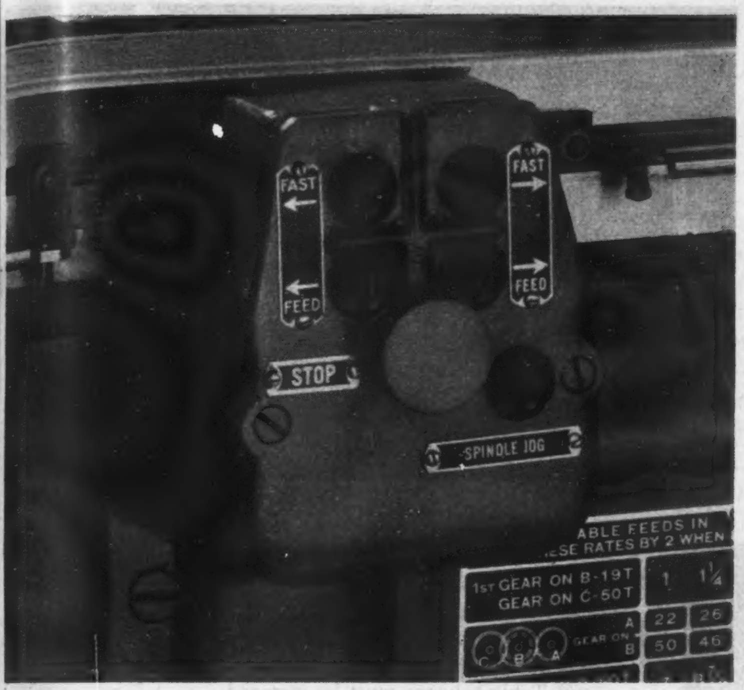


Fig. 5 — Above — Push-buttons should be so located on panels as to clearly indicate the directions of the motions they are used to control

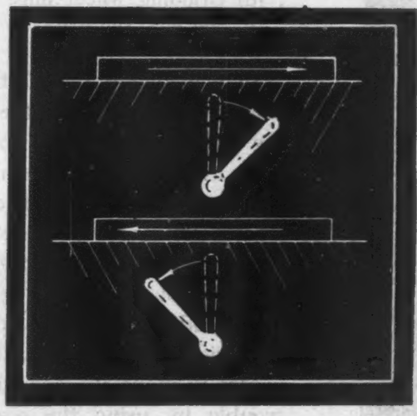


Fig. 6 — Right — Levers should throw in the same direction as the motions they produce or control

cutters, grinding wheels, revolving headstocks and index gearing. The direction of a control movement should be such that overtravel will not take hands into danger zones.

Fatigue: To keep operator fatigue at an absolute minimum, a maximum of fifteen pounds should be prescribed for all hand forces. Where controls are operated frequently this limit should be reduced to ten pounds, and in cases where small increments of movement are to be made, hand forces should be held to five pounds.

Use of Cranks: Cranks should be used where:

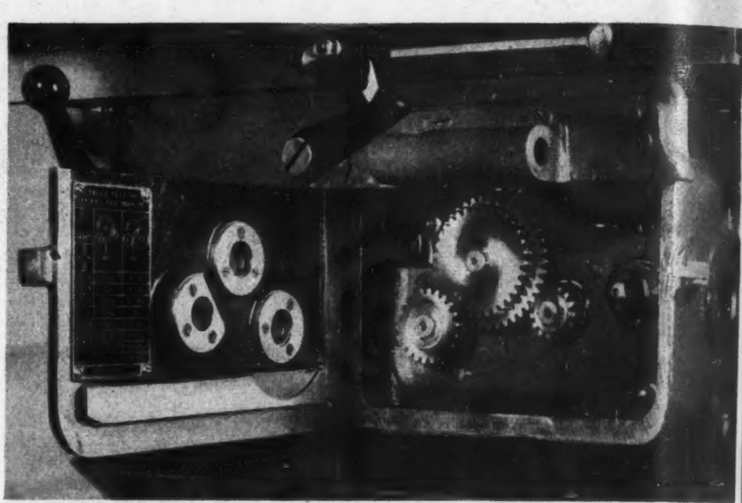
(a) A handwheel would be too large (over 18 inches diameter)
(b) A 360-degree sweep cannot be obtained (vise on machine table)
(c) Control member is to be removed after adjustment is made
(d) Rim of handwheel would project above plane of a working surface
(e) Limited use does not justify the added cost of a handwheel.

Use of Handwheels: Where it is desirable to have position of hand grip on a control unchanged by rotation of the control, use a handwheel. Some portion of its rim is always in working position regardless of the degree of rotation. Handwheels are ideal also where extremely fine increments of adjustment must be made, because they can be gripped firmly by two hands for control.

Reference to *Fig.* 1 will show how differentiation between four types of hand controls can be accomplished. In the milling machine

Fig. 7—Left—Crossfeed handwheel on this machine can be used to control headstock, coolant flow and table traverse in addition to feeding grinding wheel into the work

Fig. 8—Right—Switch in lower right corner of compartment cuts out power when door is open



pictured, speeds and feeds are changed by cranks having knurled handles. Hand movements to knee, saddle and spindle sleeve are made through handwheels. Levers having elongated or egg-shaped ends are used for clamping, while levers with ball knobs control power engagements. Incidentally, just as there is a natural tendency for most of us to "turn to the right", so is there a predominant tendency to pull a lever to clamp and push it to release.

Machines should provide toe room if the operator is required to handle the machine

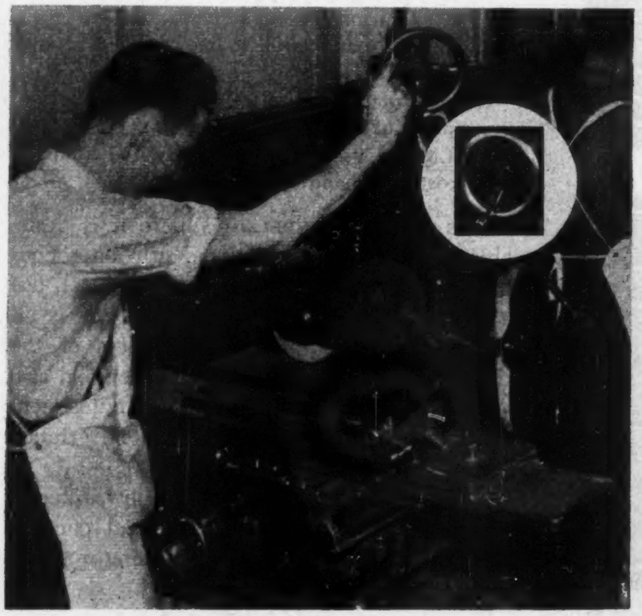


Fig. 9—Spoked handwheel is ideal control for making fine adjustments. Balanced handwheel shown in insert proved unsatisfactory because web is a barrier to the fingers

from a standing position. If controls are in or in back of the plane of the front wall of the base, the base should be undercut for a few inches above the floor level. On the other hand, if the controls project from the front wall of the machine, toe room for the operator usually is adequate.

Considerable leeway must be allowed in the location of setup controls. They should not congest the work-piece control zone but beyond this they can be high or low and in the front or back of the machine. In fact change gears, reversing switches and similar setup members usually have their positions dictated by the available space in the machine or by the positions of the power drives. A designer must, however, know the general floor plan likely to be used for a battery of his machines and guard against inaccessibility caused by back-to-back or angular arrangements of machines.

By using electric and hydraulic control circuits, designers can combine many controls on their machines and thereby greatly simplify operation. A plain cylindrical grinding machine normally requires the operation of four controls, once the work-piece has been placed between centers:

(a) Headstock must be started
(b) Flow of coolant must begin
(c) Table traverse must be engaged
(d) Grinding wheel must be brought forward into contact with the work.

In a machine such as that of *Fig.* 7, all these actions can be controlled by the crossfeed handwheel. When the work is completed, a part turn of the handwheel stops all of these actions and in addition brakes the headstock to rest. Another example is a milling machine, where the pressing of a single button will start the machine cycle by starting cutter spindle rotation, starting the flow of coolant, and engaging fast travel to move the table from loading to cutting position.

Designers have worked out many ingenious arrangements to make it impossible for the operator to get into trouble by using the wrong control. For example, in cases where power movement of a table or slide must not occur when the member is clamped, the controls can be mechanically interlocked to prevent power engagement.

Whirling handwheels are dangerous and operators usually are trained to disengage a handwheel before starting power movement. Many modern machines have automatic kick out arrangements, so that the initial or approach movement of a power lever disengages the related handwheel.

On production milling machines table dogs can be arranged to thwart unwanted actions. For example, an operator presses a button to start the fast travel movement of the table from the loading to the cutting position. But if he presses the button for movement in the opposite direction there is no response. Table dogs, in action at the loading position, hold open this unwanted control circuit. Similarly when cutting at feed rate, dogs can make it impossible to engage fast travel in the same direction.

A common setup operation is the changing of feed or speed pick-off gears. When changing or adjusting gears an operator wants to be sure no one will start his machine. On an electrically controlled machine a safeguard is readily provided which renders the machine dead whenever a change-gear compartment door is open. The pin in the lower right-hand corner of the change-gear compartment

of *Fig.* 8 must be depressed by a closed door before it is possible to complete the control circuits of the machine. No one can start the machine while the operator is handling gears and the operator cannot operate his machine without having the gears properly housed or covered.

Common exceptions to the clockwise or "turn to the right" rule mentioned earlier are the crossfeed handwheels on plain grinding machines where counterclockwise rotation brings the grinding wheel into contact with the work. This has been the practice for many years and seems to have been adopted in the belief that an operator has better control in pulling toward his body than in pushing away from it. A pull on the rim of a handwheel located at the operator's right produces counterclockwise rotation. However, that this is not an all-important reason would be indicated by the fact that thread-grinding machines of recent design use clockwise rotation to bring the wheel to the work.

The spoked slide-elevating handwheel of *Fig.* 9 has been used for many years. In a redesign the balanced handwheel shown in the insert was adopted. There was immediate complaint. Operators had for years been wrapping their fingers around the rim of the handwheel and by a slight wrist movement turning the handwheel in increments of .001-inch. The web on the new handwheel was a barrier to fingers and was removed in later designs.

Actual specifications for limiting hand forces are apt to stimulate argument. It is, however, true that the lower such forces are, the better operators like machines and the more readily they are accepted for operation by women.

Illustrative of the demand for low operating forces are the cases—embarrassing to designers and surprising to shop supervisors—where workmen insist they would prefer to operate their old machines rather than newly purchased models. Old machines, with their ways worn smooth by use, have low friction values. New machines with their accurately finished, scraped ways have higher friction coefficients. The difference is natural but the operator's reaction to it emphasizes the importance of keeping hand forces low.

Whenever an operator is trying to move a slide by small increments, or is taking off the last tenth of a thousandth, handwheel forces must be low. The force to start motion or to break a slide free is always greater than the force required to maintain motion. If the hand force to start motion is high, there is a tendency for the slide to jump ahead before the operator can reduce his hand pressure to a lower or free-motion value. (One needs but to observe what happens when the rope breaks in a tug of war to recognize how difficult it is to bring manual forces under quick control.)

Not only the amount but the type of force is important. In general, pressing a pushbutton is easy but with the present desire for long finger nails, women operators find flush-type pushbuttons troublesome. Even though forces are under five pounds they are too much for nail tips. *Fig.* 10 shows a design adopted so that a lever might be pulled to press a start button. In this the same withdrawal motion of a hand from the work fixture can trip the lever for starting the cycle. By requiring a pull for starting, the lever is made inactive when pressed or when accidental leaning against the machine occurs.

A treadle switch incorporated in the basic design of a milling machine is shown in *Fig.* 11. The switch is tripped by foot each time the operator wishes to begin the work cycle and have the machine table move from the loading position to the cutting position. This type of switch is particularly useful when an operator is using both hands in unloading and loading the work fixture. In this specific design the setup switch just above the treadle permits the treadle to start motion to the right or left, depending on the loading position selected, or can make the treadle inoperative.

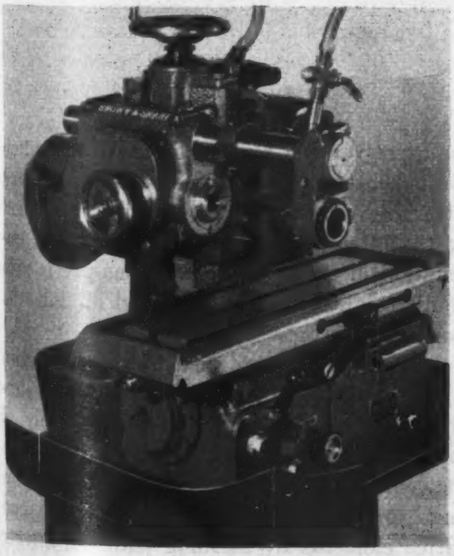


Fig. 10—Left—Showing how a pushbutton can be pressed by a lever to make operation more convenient, especially for women

Fig. 11 — Right — Treadle switches often are valuable where the operator is required to use both hands for loading and unloading work in a machine

As has been indicated, the design of all machines necessitates many compromises and few machines will conform to all the general rules herein discussed. Sometimes the price of a machine will not permit including the cost of transfer linkages often required to locate a control in an ideal spot. At other times adherence to one principle entails violation of another. In all cases, however, the basic controlling rule should be that good reasons must be presented before deviation from the ideal is permitted.

HELLCAT—World's Greatest Tank Buster

ENEMY tank men already have felt the lethal sting of the Hellcat, one of American industry's latest miracles of engineering and production.

Designed and manufactured by Buick motor division of General Motors, this high-speed, heavy fire power tank destroyer—known by the War Department as the 76-MM Gun Motor Carriage M-18—can buzz over a level concrete road at 55 miles per hour in spite of its 19 tons of weight. It mounts a 76-millimeter cannon capable of knocking out tanks point blank at several thousand yards.

Responsible for this amazing vehicle's ability to navigate at high speeds even over rough terrain is the suspension system employed. Independently sprung track wheels with torsion bars are used instead of conventional coil or volute springs. In addition, heavy-duty automotive-type shock absorbers provide cushioning against road shocks.

In developing the Hellcat's transmission, engineers drew unstintingly from their vast backlog of "know how" gained throughout many years of automotive design. They came up with a power train composed of a final drive coupled with the GMC Torqmatic transmission which makes it possible for the operator to shift easily under full load with surprisingly few hand motions. The power plant is a nine-cylinder, single-row, radial, aircraft-type engine developing 480 horsepower and mounted so as to be readily removable even under adverse conditions.

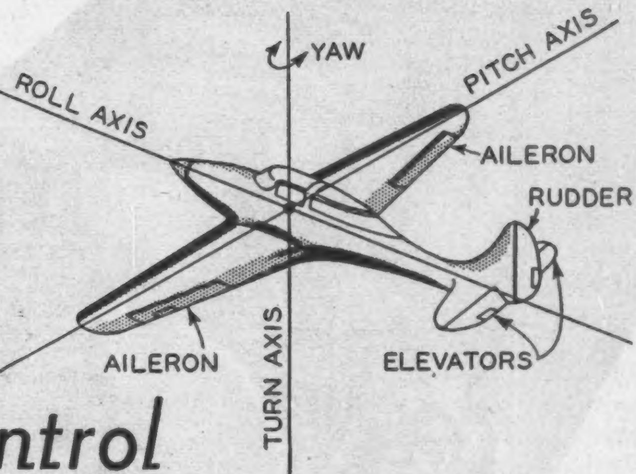
Design of the cooling system was accorded special attention because these vehicles must operate without hitch under the severe heat conditions encountered in desert and tropical warfare. Centrifugal blowers are utilized for air cooling the oil of both torque converter and engine. For operation in cooler climates, a damper is provided in the outlet air duct to direct heated air into the fighting compartment for crew comfort.

Guiding of the track is accomplished by lugs forged in the track links to mesh with the track wheels and support rollers. The track itself comprises an assembly of 83 heavy drop-forged links proofed against wear by a heavy layer of wear-resistant alloy. Long track link pins are used. These project from each side of the links and serve as the drive elements by meshing with the driving sprocket teeth.

Despite the use of a 76-millimeter cannon, requirements of speed and maneuverability demanded weight reduction at every turn. This entailed the development of a new type of light turret mechanism and an advanced type of gun mount. The light weight turret presented a new problem in turret balance which was solved by adding a stowage box to the rear of the turret to balance the weight of the gun in the trunnions.

Contract for the Hellcat was issued in January of 1943. Ordnance required delivery of the first pilot model by April with production deliveries beginning in July. Thus, pilot models had to be built practically by hand while preparation for production development was going on.

Electronics Sets Standard for Automatic Control



By W. H. Gille and R. J. Kutzler
Minneapolis-Honeywell Regulator Co.

PROBABLY no other control problem offers as many opportunities for the advantageous application of electronics as the design of equipment for automatic flight control. The Autopilot used in the Army's heavy bombardment aircraft is typical of equipment in which electronic devices have produced a flexible, coordinated system from what was formerly an unwieldy electromechanical machine.

Control of an airplane can be resolved about three perpendicular axes, as shown in the head illustration. When the position of the airplane is such that its roll and pitch axes are horizontal and its turn axis is vertical, it will fly "straight-and-level"; but whenever the airplane is made to rotate about one or more of these three axes by movement of its ailerons, elevator or rudder its flight path is changed. Position of these control surfaces does not determine the airplane's position, but rather the rate of the plane's rotation about its corresponding axes. This characteristic of airplane flight control makes necessary double operations in executing simple maneuvers.

Another complication in aircraft flight control arises from the fact that, in most turning maneuvers, the ailerons, rudder and elevator must all be operated simultaneously to produce a proper turn. Too much rudder will cause a skid; too much aileron, a slip; and too little up-elevator will permit the plane to lose altitude in the turn. Therefore, any automatic pilot must accurately proportion the movement of each control to fit a complex pattern, and, at the same time, coordinate the movement of all controls to maintain them in proper relationship with one another.

In addition, it is desirable for an automatic pilot to be so universally adaptable that identical units from the production line will operate effectively when installed in different types of airplanes. Since different types of aircraft have widely different flight characteristics, and since these same characteristics are influenced materially by speed, loading, altitude, and weather conditions, an automatic pilot, to be universally adaptable, must have pro-

Fig. 1—Right—Gyro for vertical flight control

From a paper presented before the Los Angeles meeting of the A.I.E.E., Aug. 29 to Sept. 1, 1944.

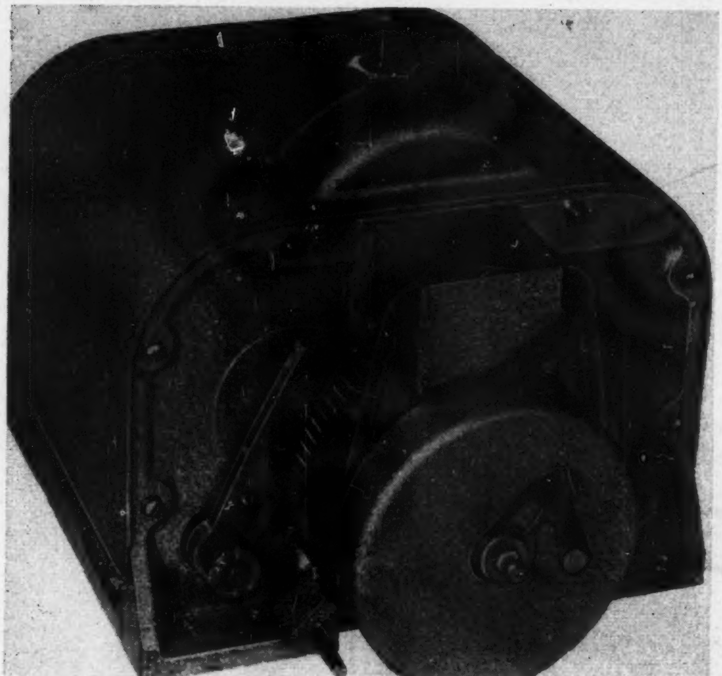


Fig. 2—Above—Directional stabilizer employs a horizontal gyroscope and potentiometers

Fig. 3—Right—Schematic diagram for automatic-pilot control system

visions for easily adjusting its operating characteristics to match those of the airplane. The use of vacuum tube circuits not only simplifies the solution of these problems but also makes possible a degree of sensitivity and accuracy not obtainable by other means.

In the Autopilot the vertical flight gyro, *Fig.* 1, is employed as a reference for roll-axis and pitch-axis deviations. Its outer case is mounted firmly to the structure of the airplane, but its rotor is universally mounted within the case so that the gyro's automatic erection system can maintain the rotor axis vertical, regardless of the position of the airplane. Any tilting of the gyro case with respect to the rotor is measured electrically by four potentiometers within the unit. These potentiometers have their windings fastened to the case and their sliders held in position by the gyro rotor. The resulting electrical signals activate the system to restore the aircraft to level flight. Since pitch-axis deviations are corrected by elevator control alone, only one potentiometer is located on the pitch axes. However, three potentiometers are located on the roll axis of the unit, as all three controls are employed to correct roll-axis deviations.

Directional or turn-axis control is derived from the directional stabilizer, *Fig.* 2, which contains a horizontal gyroscope. The rotor of this gyro tends to remain fixed in space, unaffected by turning movements of the airplane. Here again, potentiometers with their windings fastened to the outer case and their sliders held in position by the gyro rotor assembly are used to measure electrically any course deviation of the airplane. Two potentiometers are used, one for rudder and the other for aileron control.

Thus, in the two gyro-controlled units, a total of six potentiometers are employed as the sensing elements of the system. The individual units are identified in *Fig.* 3 by the numbers 1 and 10, and the potentiometers by the numbers 5, 6, 11, 12, 13, and 14. A seventh potentiometer, No. 2 in the illustration, is an auxiliary unit to energize a direction indicator, No. 7, on the pilot's instrument panel.

No less important in operation are the motorized servo units, *Fig.* 4, which do the actual work of moving the control surfaces. Three such units are required, one each for ailerons, rudder and elevator, as shown in *Fig.* 3 at Nos. 15, 18 and 19, respectively. Each servo unit contains an electric motor driving a cable drum through a differential gear train. Two solenoid-operated clutches control the direction of cable-drum rotation. An automatic brake mechanism effectively holds the cable drum stationary when neither of the two clutch solenoids is energized. Cables wound on the servo cable drum are attached to the main control cables which move the control surfaces.

Each servo unit also contains a balancing potentiometer with its winding fastened to the servo-unit frame and its slider attached to the cable drum. The purpose of these balancing potentiometers will be revealed later in the discussion of the controlling bridge circuits.

The 7-tube amplifier, *Fig.* 5, contains three distinct channels, one for each of the three flight control axes, *Fig.* 3, No. 16. Each channel contains two tubes—a 7F7 double amplifier stage, a 7N7 discriminator stage—and two relays which operate the clutch solenoids of the re-

lated servo unit. For the moment, each channel of the amplifier may be thought of as a motor-controller which analyzes the amount and direction of control-surface movement called for by the sensing potentiometers in the bridge circuit and operates the corresponding servo unit accordingly.

Manual control and adjustment of the system are provided for by the control panel, *Fig.* 6. This unit centralizes at the pilot's fingertips the various switches and control knobs by which he engages the system, aligns the Autopilot with a selected attitude and heading, adjusts the sensitivity of the Autopilot, regulates the amount of control surface applied to correct a given deviation, and coordinates interrelated control functions. A "turn control" is included by which the human pilot can execute turns and evasive tactics without disengaging the Autopilot.

Power for the direct-current components of the system is supplied by the airplane's batteries. A rotary inverter provides the 19-volt, 105-cycle alternating current power required. The entire equipment—7 motors, 14 solenoids, and 7-tube amplifier energized and operating—draws only 7.6 amperes from the 26-volt batteries. Normal voltage and frequency variations have relatively little effect on performance.

If the gyroscopes are thought of as the *eyes* of the

Fig. 4—One of three servo units for control surfaces

Autopilot, the servo units as the *muscles*, and the amplifier as the *brain*, then the bridge circuits might be likened to the nervous system by which all these components are interconnected. There are three distinct bridge circuits of the resistance type in the system, one each for aileron, rudder and elevator control. Each bridge circuit provides the input to a specific amplifier channel which, in turn, operates one of the servo units.

In *Fig.* 7 is shown a simplified schematic diagram of the aileron bridge circuit. The control potentiometer is located in the vertical flight gyro and is sensitive to roll-axis deviations of the airplane. The balancing potentiometer is actuated by the cable drum on the aileron servo unit. Both potentiometers are energized by a common transformer in the amplifier. As long as e_1 and e_2 are equal, E is zero and no signal is applied to the amplifier, and the servo unit remains stationary; but when the control potentiometer is displaced, e_1 and e_2 are no longer equal, and a signal E is impressed on the amplifier. Depending on the phase characteristic of E, the amplifier then causes the servo unit to drive the balancing potentiometer in the proper direction to make e_2 equal to e_1 and thereby reduce E to zero. In effect, the potentiometers, amplifier, and servo unit comprise a self-balancing bridge circuit in which the balancing potentiometer produces a corresponding movement of the cable drum to drive the control surfaces of the airplane.

This is the fundamental operating principle of the Autopilot. Actually, of course, the circuit is complicated somewhat by the necessary inclusion of additional control potentiometers, as shown in the diagram of the aileron

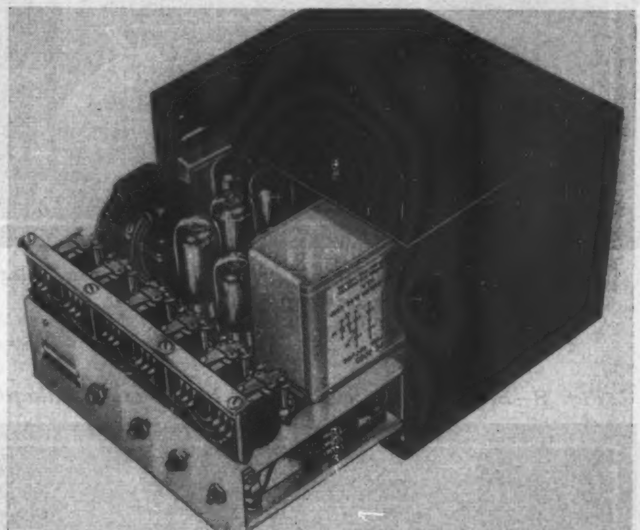
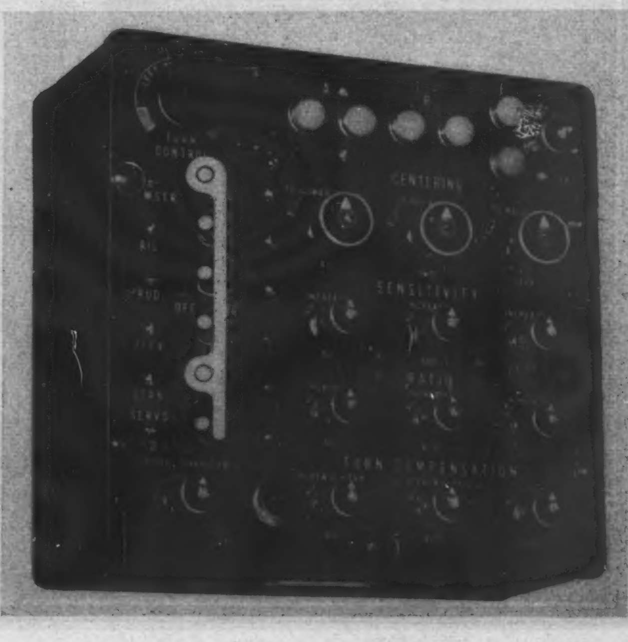


Fig. 5—Above—Amplifier unit for automatic pilot

Fig. 6—Below—Control panel for manual adjustment and coordination of automatic pilot



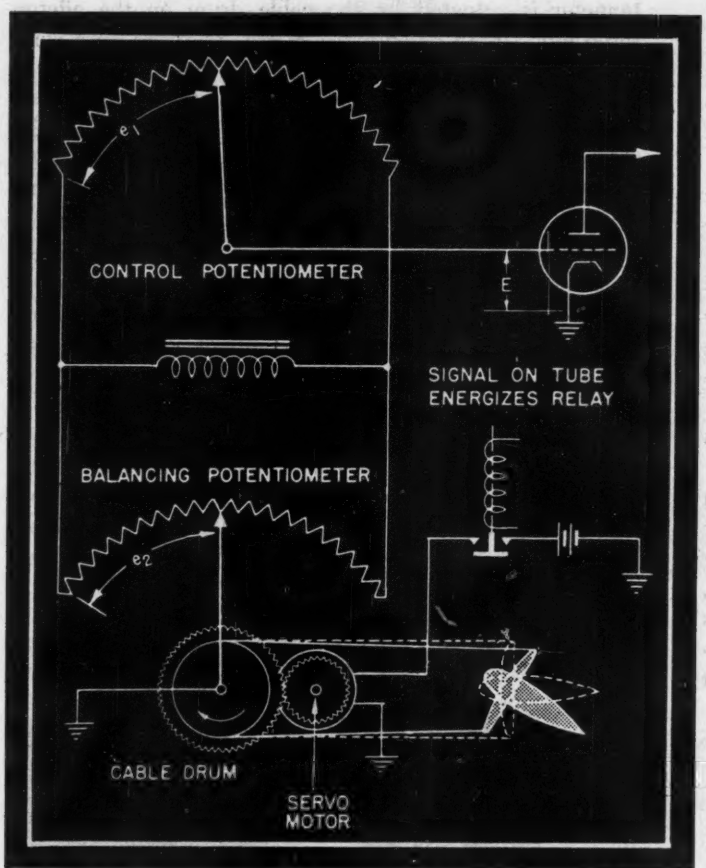
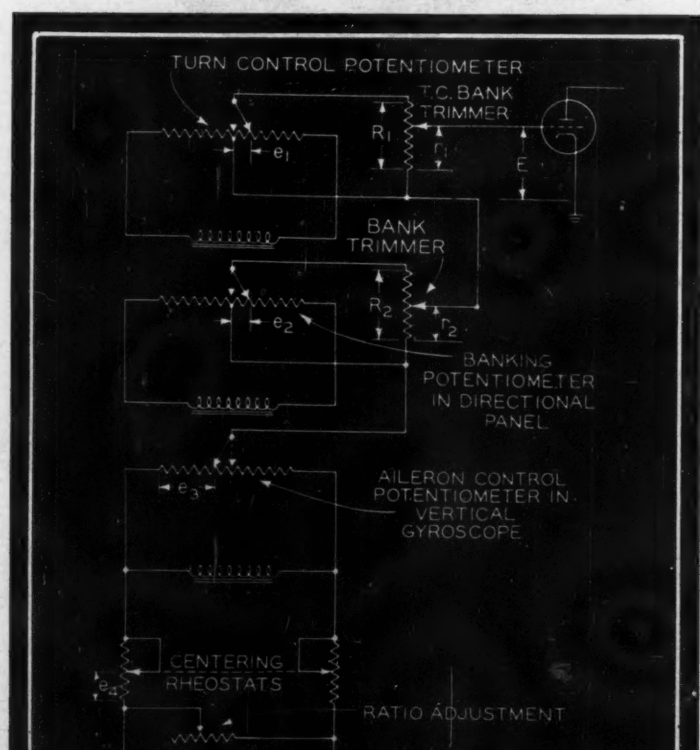


Fig. 7—Above—Simplified bridge circuit for aileron control shows basic principle of operation

Fig. 8—Below—Complete aileron bridge-circuit diagram



bridge circuit, *Fig.* 8. The principle of operation, however, is the same.

Among the advantages of using potentiometers in alternating-current bridges are:

1. Simple isolation of voltage sources
2. Controllable amplification of small signals
3. Linearity of control characteristics
4. Nondistortion of wave forms
5. Full 180-degree phase change at balance point.

A linear relationship exists between the deviation in attitude of the plane, as measured by the potentiometers on the gyros, and the displacement of the control surfaces. The voltage output of a bridge potentiometer can be considered as practically proportional to the displacement of the sliders, because of the high impedance of the amplifier input circuit when compared to that of the bridge circuit. The one exception to this linearity is the banking potentiometer on the directional gyro. Represented by a single potentiometer with linear characteristics in *Fig.* 8, it is actually a combination of two potentiometers. These two potentiometers are so arranged that their sliders are displaced in unison, *Fig.* 9, and the resulting voltage e_2 is a parabolic function of the turn angle θ.

Bridge Simplifies Control

Employment of the resistance bridge with electronic motor control permits the substitution of simple and easily accessible electrical adjustments in place of more difficult mechanical adjustments. As an analogy, imagine how much easier it would be to adjust the carburetor, tappets and distributor of your automobile if it could be done by turning knobs as you do in selecting a station on your radio. In the electronic pilot this case of adjustment is attained by simply connecting rheostats, located on the control panel, to the bridge circuit as shown in *Fig.* 8. Two centering rheostats are mounted on a common shaft connected in series with the balancing potentiometers. When the resistance of one is increased, the resistance of the other is decreased. This arrangement permits the pilot to shift the electrical center of the balancing potentiometer in either direction to align the circuit with the mechanical trim position of the control surfaces. These centering units may be likened to "electrical trim tabs".

Another rheostat connected in parallel with the balancing potentiometer regulates the voltage drop across the potentiometer winding. This "ratio control" governs the amount of control surface movement produced by a given displacement of the control potentiometer. This adjustment might be considered an electrical "gearshift", as it varies the airplane's speed of response for a given deviation. With slight variations, the rudder and elevator bridge circuits are similar to the aileron circuit just described.

So far, the amplifier has been considered merely as a phase-sensitive motor controller which responds to any unbalance of the bridge circuits by closing the proper relay, thereby producing the required

servo-unit operation. How the amplifier works and how it introduces special control characteristics are revealed by the schematic diagram of one of its three channels, *Fig.* 10. As mentioned previously only two tubes are used in each channel, one 7F7 and one 7N7. The first section of the 7F7 tube is used strictly as an amplifier, with no direct-current bias applied to the grid. The second section, however, is employed in a unique way as a control tube, with three separate controlling voltages applied to its grid.

Attention is called, first, to the resistance *R* connected in series with the grid of the second section. It is apparent that any voltage impressed on the left-hand end of this resistor is proportioned between the resistor and the internal grid resistance. When the applied voltage is negative, no current will flow between cathode and grid, as the grid resistance becomes practically infinite. Therefore, the entire applied voltage is impressed on the grid. However, when the applied voltage is positive, current flows between cathode and grid, because the grid resistance becomes low in comparison with the resistance of *R*, and hence only a small fraction of the applied voltage is actually impressed on the grid. Thus, negative signal voltages applied to the resistor produce sharp plate-current fluctuations, whereas positive voltages do not cause any appreciable change in plate current, *Fig.* 11. In effect, this resistor lowers the saturation point of the tube's characteristic curve and produces a very flat curve beyond that point.

It is apparent from the resulting characteristic curve, *Fig.* 11, that the *sensitivity* of the tube can be controlled by applying a variable positive bias. Thus, when the tube is biased at point *A* (high sensitivity) a small signal will produce plate-current variations. But when the tube is biased at point *B* (low sensitivity) the same signal is ineffective, and a larger signal is required to produce plate-current fluctuations. This sensitivity bias is knob-controlled by a potentiometer located on the control panel.

The sensitivity bias is applied continuously but, under certain conditions of operation, two additional bias voltages are applied to this same tube section. One is a negative feed-back voltage which has the effect of increasing the tube's sensitivity whenever either half of the 7N7 discriminator tube is passing current. The other is a throttle voltage, of higher positive value than the sensi-

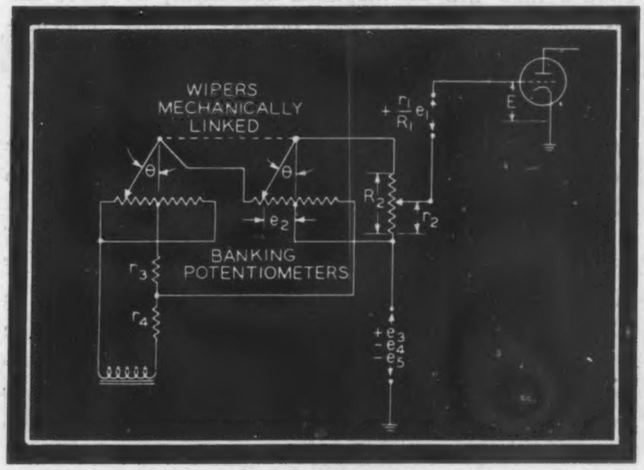
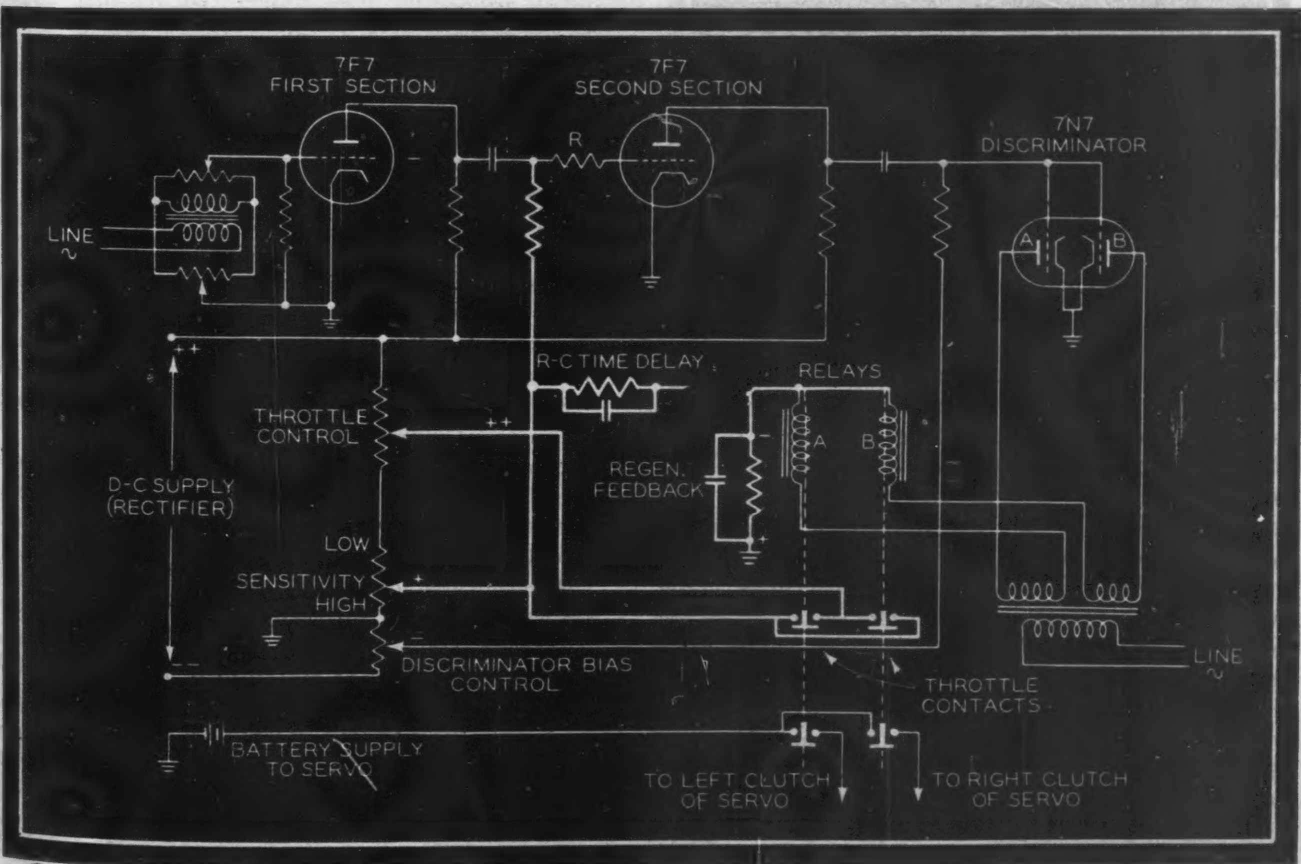


Fig. 9—Above—Partial circuit showing dual potentiometer arrangement of banking potentiometer in Fig. 8

Fig. 10—Below—Schematic diagram for one amplifier channel shows unusual method of control



tivity bias, which is applied to the tube whenever the throttle contact on either of the two relays is closed. The effect of this throttle voltage is to increase the bias on the grid to point *C*, *Fig.* 12, thus blocking any signals which are not sufficiently large to overcome this increased bias. A signal which produces plate-current fluctuations when the tube is biased at *B* is ineffective when the bias is changed to *C*. A resistance-capacitance time-delay circuit serves to maintain the throttle bias on the tube for a short time after the relay contacts reopen.

The purpose of the feedback circuit is to insure that the weakest signal capable of overriding the bias of the control tube will be amplified sufficiently to drive the discriminator tube, and thus produce positive, nonchattering operation of the relay.

The discriminator tube, which employs alternating-current voltage on the plates, is biased negatively to the cut-off point. This bias is sufficiently large to prevent any current flow through either half of the tube when no signal is applied to the grid.

It is apparent from *Fig.* 10 that since the two plates of the discriminator tube are connected to opposite ends of similar transformer windings, the voltage of one plate is always 180 degrees out of phase with the voltage of the other plate. When the signal applied to the interconnected grids is positive, the electrons will flow from the cathode to whichever plate is positive at the same instant.

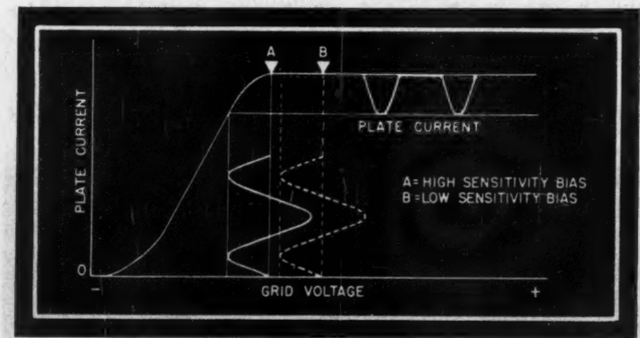


Fig. 11—Effect of changing sensitivity bias of control tube

Since each plate circuit contains a relay coil, the corresponding relay is energized. Thus the discriminator tube operates one relay or the other, depending on the phase relationship between the input signal and the alternating-current plate-voltage supply. In effect, this discriminator is a grid-controlled rectifier similar to a thyratron.

As illustrated by the diagram, individual relays control the operation of separate clutches in the servo unit, and thus control the direction and amount of servo-unit rotation or control surface movement.

Now, let's follow a signal from the bridge circuit all the way through the amplifier circuit. This signal will be one which calls for counterclockwise rotation of the servo unit, as would be produced by energizing the left-hand clutch solenoid. That is, the signal is in phase with the voltage at plate *A* of the discriminator tube. The signal is first amplified and reversed in phase by the first section of the 7F7; then it is applied to the grid of the second section where it produces plate-current pulsations during the negative half of the cycle. Since the signal phase is again reversed in the second section, these plate current pulsations apply a positive half-wave signal to the discriminator tube which is in phase with plate *A* of that tube. The resulting current flow energizes relay *A*, which in turn energizes the left-hand clutch solenoid of the servo unit.

At first the signal on the control stage grid of the 7F7 is sufficiently large to override the additional positive bias introduced by the closure of the throttle contacts on the relay. However, as the servo unit drives the balancing potentiometer toward the balance point, the signal is gradually reduced until its negative peaks no longer exceed the throttle bias. When this occurs the signal to the discriminator tube is interrupted, causing the relay to drop out for about ⅛-second while the time-delay condenser discharges.

When the condenser is discharged, the throttle bias is

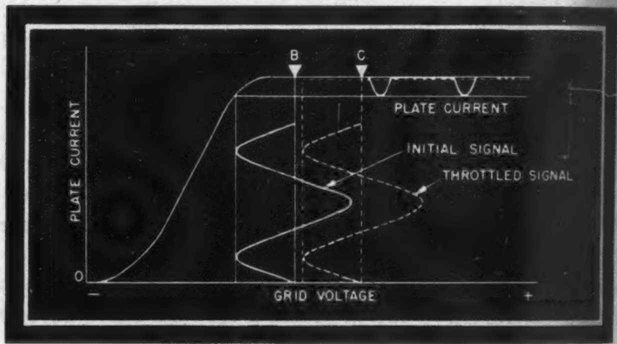


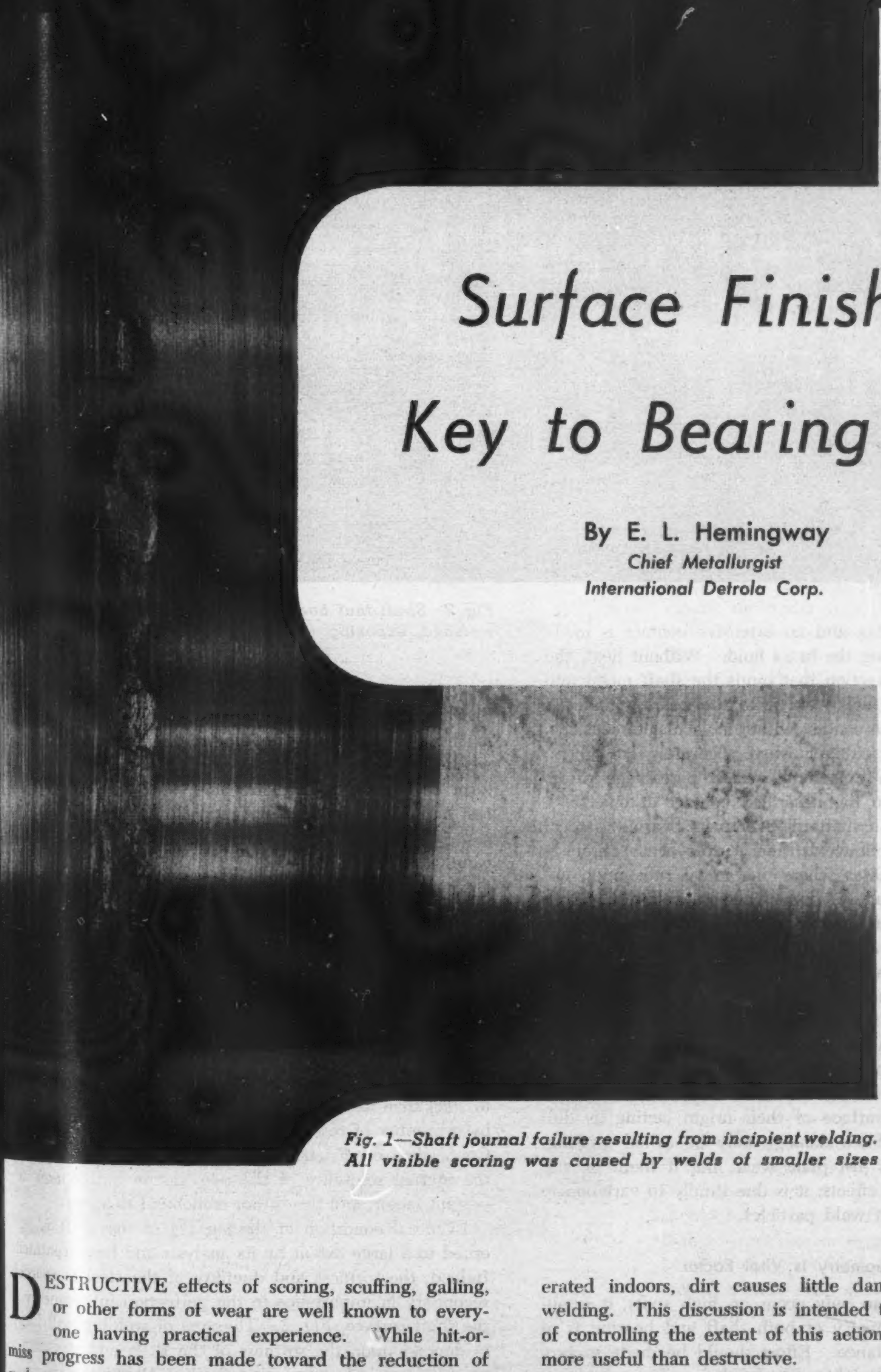
Fig. 12—Curve for control tube illustrating effect of throttling bias

lost and the signal to the discriminator is restored. The relay closes again, but only momentarily because the throttle bias is thereby reapplied. The result is that, as the servo unit approaches the balance point, it operates intermittently with a "pecking" action which prevents overshooting or overcontrol of the airplane. The rate of correction, which is fast for large deviations, will gradually diminish as the airplane approaches a stabilized flight attitude.

In automatic flight control, as in other control problems, application of electronic control principles has made it possible to set new standards and to achieve full flexibility of control with completely automatic operation of all control functions. Furthermore, these advantages have been obtained with actual reductions in equipment weight.

Since the use of an automatic pilot for precision bombing imposes complex and rigid requirements which are not encountered in commercial aeronautics, it can be confidently stated that the continued development of electronic controls will produce a small, lightweight, foolproof autopilot for peace-time use.

Although the connection now seems obscure, some of the applications in the development of the Autopilot were taken from a cabin temperature control system. In a similar way, the principles employed in both systems have been found useful in the automatic regulation of engine power. And so, in a seemingly endless chain, each new development brings to light the solution of another problem. Who knows what application these same developments will find in enlarging the prosperity, health, and happiness of a peaceful world?



Surface Finish—Key to Bearing Life

By E. L. Hemingway
Chief Metallurgist
International Detrola Corp.

Fig. 1—Shaft journal failure resulting from incipient welding. All visible scoring was caused by welds of smaller sizes

DESTRUCTIVE effects of scoring, scuffing, galling, or other forms of wear are well known to everyone having practical experience. While hit-or-miss progress has been made toward the reduction of such wear, the basic causes do not seem to be getting the proper consideration. It is believed that, if they were, improvements in plain bearings (as well as in all other rubbing metallic surfaces) would be accelerated.

Two conditions most damaging to plain bearings are dirty or inefficient lubrication and minute high spots on the shaft and its bearing which cause welding as in *Fig.* 1. In mechanisms such as internal combustion engines or farm machinery, dirt and dust undoubtedly cause considerable abrasive wear but, in the average machine operated indoors, dirt causes little damage compared to welding. This discussion is intended to point out means of controlling the extent of this action and of making it more useful than destructive.

Welding occurs when high points of mating surfaces are rubbed together under sufficient pressure to wipe away all contaminating surface films (both lubricant and solid films, such as oxides, etc.), making a clean, actually metallic contact. Though the affected area will usually be minute, just as true a weld is formed as that made with a torch. Heat is present and makes such welding extend over larger areas because the metal is thus made more plastic, but melting is not necessary. In the brazing-torch technique, however, the surface is made abso-

lutely clean with flux and an extensive contact is made possible by rendering the brass fluid. Without heat, the same molecular attraction that binds the shaft metal into one integral shape will weld it to the bearing metal if both are rubbed into contact within molecular dimensions.

No bearing surface is finished accurately enough so that more than an exceedingly small percentage of its area can come into the described contact at one time. Yet the motion of a shaft relative to its bearing is ideal for the production of welds, and it does occur regularly in practically all cases where one metal rubs upon another. More damage is done by such welding to both machine bearing and cutting tool surfaces than by any other form of wear.

The described welding action is instantaneous, and continued movement of the shaft just as suddenly breaks the weld apart. As the break occurs, a particle of metal is torn from one surface to become a ragged defect upon the other. These projections may be more or less rapidly broken loose but, during their existence, they tear their way through the surface of their origin, acting as dull tools to ruin it. Such damage may be called scuffing, scoring, galling, or just plain wear, but if there be any difference in these effects, it is due simply to variation in size of the torn-out weld particles.

Geometry Is Vital Factor

Since welding can only occur on high spots, it is obvious that the geometry of both shaft and bearing is of the greatest importance. Effort should be made to produce as perfect straightness, roundness, alignment and clearance dimension as is economically possible. Control of smoothness within close limits also is necessary for reduction of welding.

Even though the utmost is done to produce good geometry, the surface will still be exceedingly mountainous compared to the size of an oil molecule. "Running-in" of the bearing assembly to satisfactory operation must then depend upon the generation of still better geometry by *controlled welding wear*. There seems to be a critical size of weld which, if not exceeded, results in the removal of high spots by so minute a process that increased accuracy in shape and greater smoothness are produced. If that weld size be exceeded, greater roughness follows in the form of scoring, galling, freezing, etc. However, there is a limited amount of metal that can be removed by any controlled welding wear, and thus dimension and shape cannot be neglected.

Fig. 2—Shaft that has been ground then partially Superfinished, exposing the pattern of high spots characteristic of ground surfaces

Kind of metals employed in the shaft and bearing are important also. Since steel is almost universally used for shaft purposes, there has been a continual search for metals or alloys, even layers of different metals, in the same bearing, that have a lower welding affinity for steel and yet have the required physical values and resistance to corrosion. All such metals have some capacity for welding but there are considerable variations in that respect. Just why one metal has a greater tendency to weld to steel than another has been the subject for speculation by a number of researchers. Ernst, at the 1940 Summer Conference on Friction at M.I.T. stated his belief that the mutual solubility of the two metals is the most important factor, and the author is inclined to agree.

Eventual condition of the surface of the shaft is governed to a large extent by its analysis and heat treatment. Indeed, the softness and ductility of the steel is scarcely secondary in importance to the perfection in geometry of the shaft surface, *Fig*. 2. The area of welding will largely depend upon the softness of the shaft and the degree of roughness of its surface finish. Obviously, a given pressure will rub and deform a soft steel ridge to a greater area of contact than it will a hard one; and, if the surface finish is coarse, there will be fewer ridges to support the load resulting in greater pressure upon each individual one. No control over welding is possible on coarse surfaces.

After the weld has formed, the depth to which the particle is torn out will depend upon the ductility of the steel. If it is hard and brittle, not only will the area be

small, but an extremely thin layer will flake off. If the steel has high ductility, however, the particle will tear out to a greater depth and project from its adopted surface to a destructive dimension. It will be hardened by cold deformation and will severely score its soft parent surface unless its size is restricted by a closely controlled scratch depth and pattern.

Tests made under the author's direction, using a Brush surface analyzer as a means of measurement, show a representative comparison of the damage done to hard and soft surfaces when loaded to the point of seizure. Two shafts, one 11 rockwell C hardness and one 62 rockwell, running against 80-10-10 bronze were used. Both were ground to 12 microinches profilometer reading. The soft shaft was scored approximately 20 times as deeply as the hardened one as shown in *Fig.* 3. Reduction in ductility by hardening is a really efficient means of restricting weld particle size.

Scratch depth and pattern also are instrumental in the success or failure of bearing operation, and much less is known about such effects than about the benefits derived from hardening. One fact that should not be ignored is that grinding ridges upon supposedly hard surfaces are always annealed to some extent by the heat generated in that process. This condition is often contributory to disappointing results from hardened shafts.

Scratches Serve Dual Purpose

Two objectives exist for the employment of scratches. One is to provide a closely spaced system of alternate load bearing ridges, and valleys containing the contaminating lubricant. The second is to produce a finely divided pattern upon which no large weld could form and which would break up the tendency for metal to accumulate ahead of the weld causing galling or freezing. Fortunately, hardened and smoothened surfaces, from which the annealed grinder ridges have been removed by a process such as Superfinish, can support heavy loads while welding remains of less than the critical size. Under normal loading the removed particle seems to be almost molecular in size and surfaces as smooth as these become still smoother. There need be no scratch system on shafts that are really hard on the outermost dimension, and of good geometry.

Breaking-in of soft steel shafts is often an entirely different matter, especially if the bearing is made of an alloy such as bronze and the shaft is heavily loaded. While soft shafts can and probably should be very smooth for light duty, it is advisable to employ a controlled scratch pattern for heavier loads. The depth of those scratches must be at least equal to the maximum amplitude of the surface waviness so they will still be present after breaking-in. Otherwise, continuous areas would be developed upon which uncontrolled welding could proceed without sufficiently minute distribution of the lubricant. Yet it is not advisable to employ a greater scratch depth than necessary because by so doing there would be fewer load-carrying ridges.

Both the author's work and that of several others indicate that soft-steel shafts should have a finish of from 5 to 12 microinches (root mean square) for anything but light duty. A finish finer than 5 microinches seldom will allow sufficient metal for complete wearing away of waviness

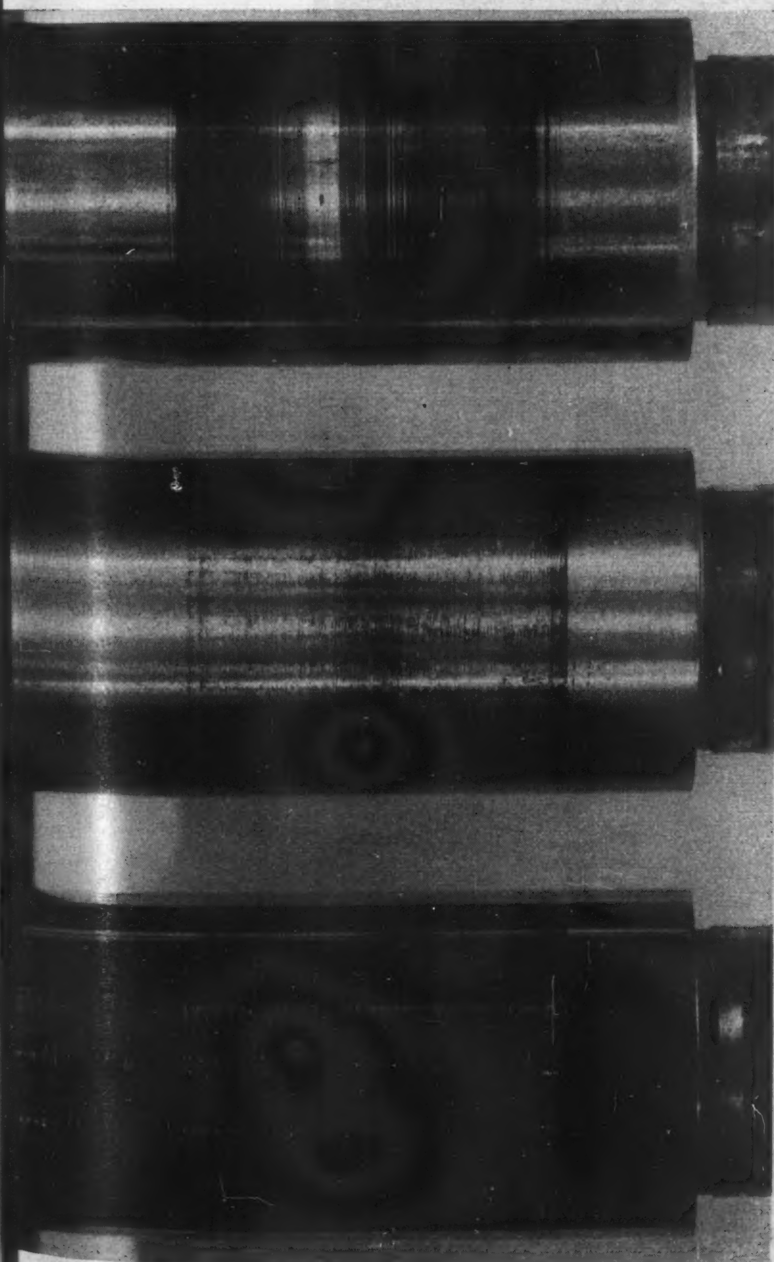


Fig. 3—Left—These test shafts have been run to seizure. Top shaft is soft steel and failed at 1400 pounds with damage twenty times as great as the middle shaft which was hardened and ground to 12 microinches and failed at 1550 pounds. Bottom shaft—hardened, ground and Superfinished—failed at 3100 pounds. Grinding defects on middle shaft have been partially removed by welding to create better geometry. Practically no damage from seizure is apparent on bottom shaft surface

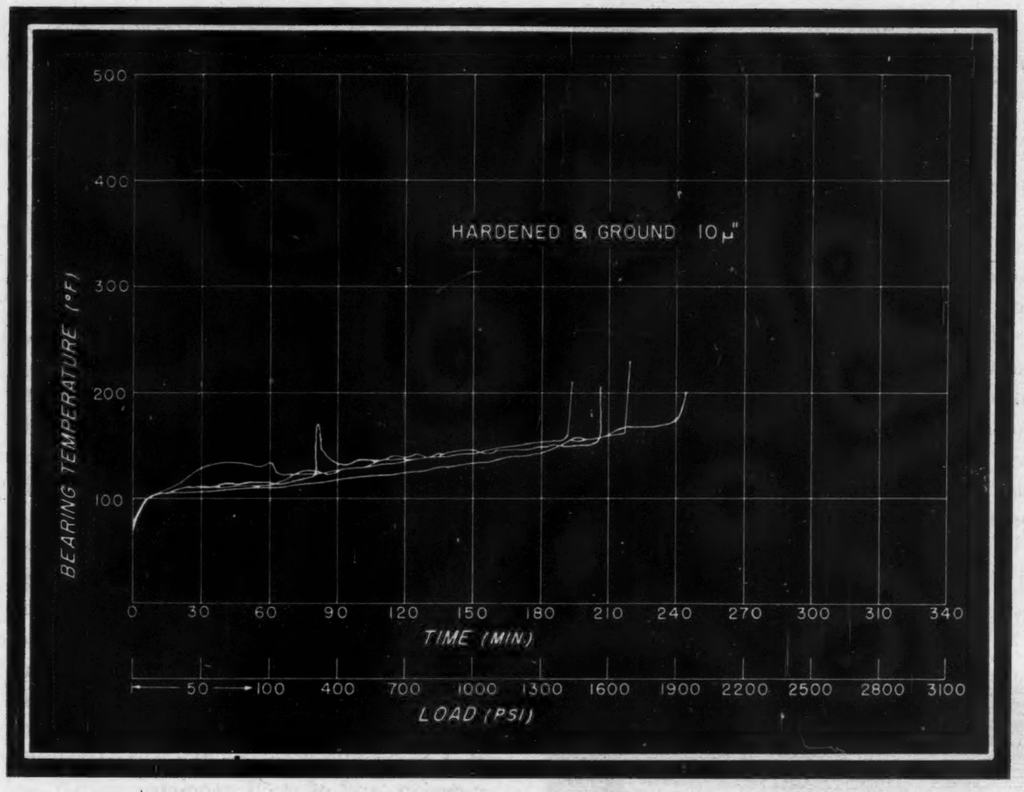


Fig. 4—Left—Four test runs on shafts hardened and ground to 10 to 12 microinches, root mean square. Average load carried 1625 pounds per square inch

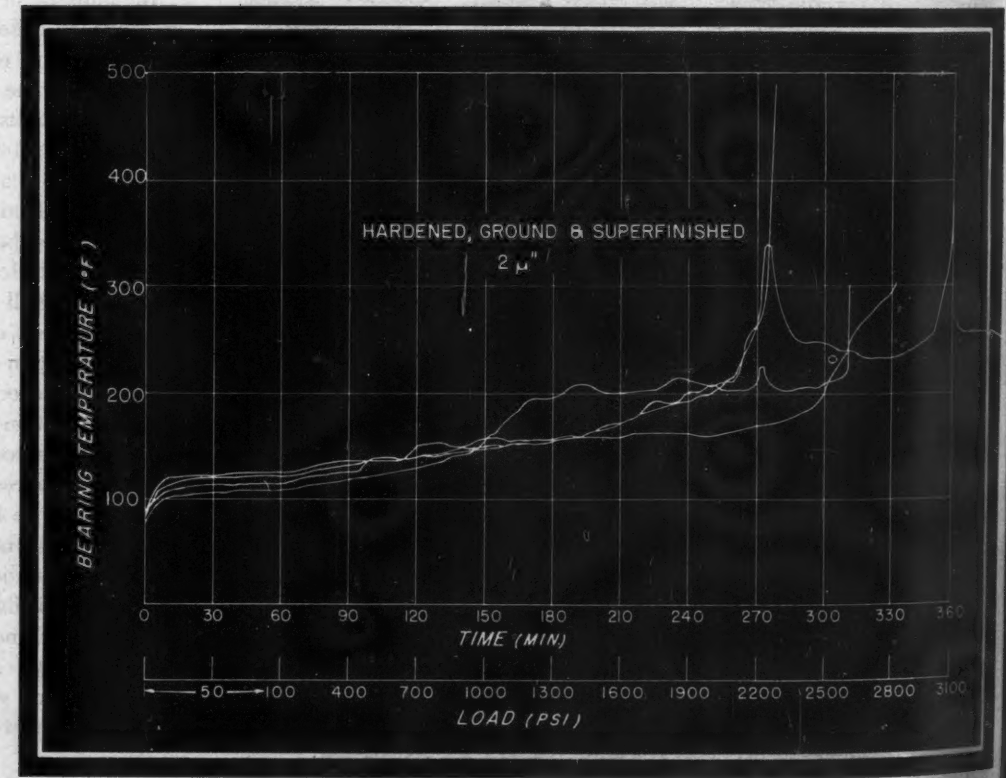


Fig. 5—Right—Four test runs on shafts hardened, ground and Superfinished to 2 microinches. Average load was 2700 pounds per square inch

without losing control of weld particle size. If the roughness is greater than 12 microinches, there will be too few ridges to carry the load, and welds above the critical dimension will be created on the first operation. Analysis of many reports of bearing tests indicate that such scratch patterns become increasingly effective as they are more uniformly and finely spaced. There should be no extended areas between scratches.

The testing machine employed is shown in *Fig. 8*. It has a bearing one inch in diameter and one inch long with load capacity of 4000 pounds per square inch of projected area. Oil is supplied at a measured rate by gravity, and bearing temperatures are measured with a Leeds and Northrup potentiometer. Speed of rotation used in the tests discussed in the following was 1730 revolutions per minute. Each test was run-in one hour under a tare load of 50 pounds. Thereafter, 50 pounds were added every 5 minutes and conditions of torque and tempera-

ture observed until seizure was indicated by stopping of the one-horsepower motor. Soft shafts were untreated X1335 steel and hard shafts were carburized and hardened X1315 steel. The bushings were 80-10-10 bronze.

This machine shows variations in torque instantly and constantly. Each welding action of any consequence is indicated by a movement or "skip" of the torque arm. It is thus possible to note the welding characteristics of each variation in hardness and surface finish. Hard and smooth surfaces run with much less skipping of the arm as compared to soft. Rough surfaces of 25 microinches or more show almost constant severe welding, heat rapidly, and fail at less than 30 per cent of the capacity of fine finishes, with considerable damage to the bushings. There was an invariable relation between severity of skipping and temperature rise. Light and continuous welding

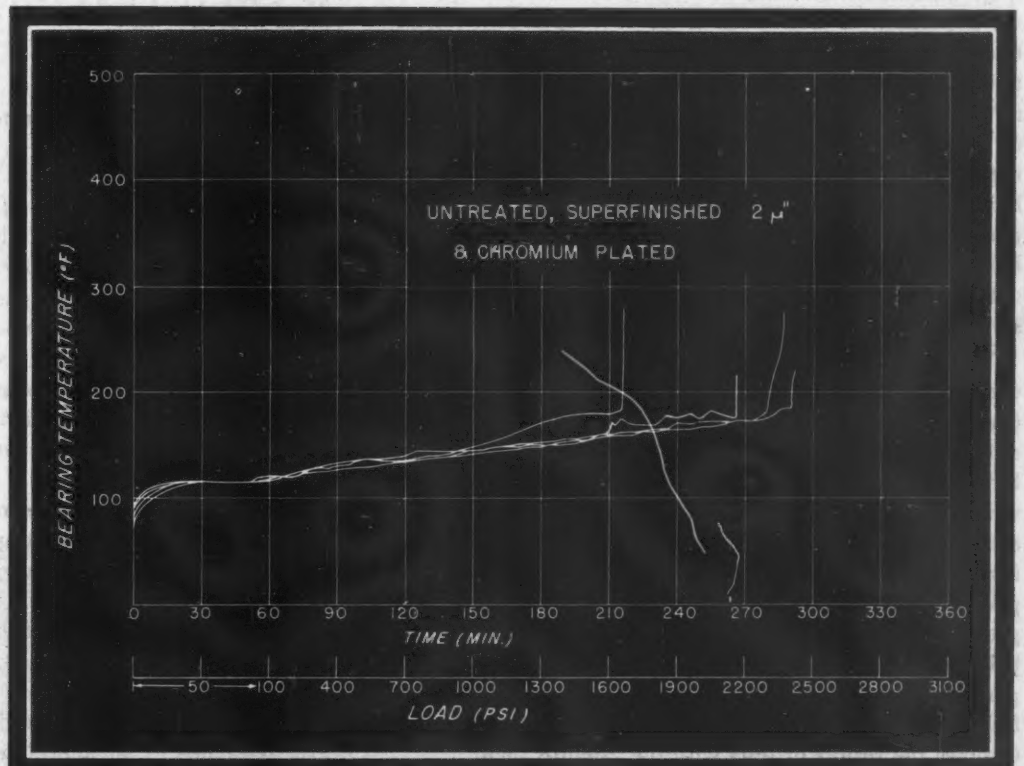


Fig. 6—Right—Test runs on soft shafts, Superfinished to 2 microinches and bright chromium plated. Average load was 2150 pounds per square inch before failure

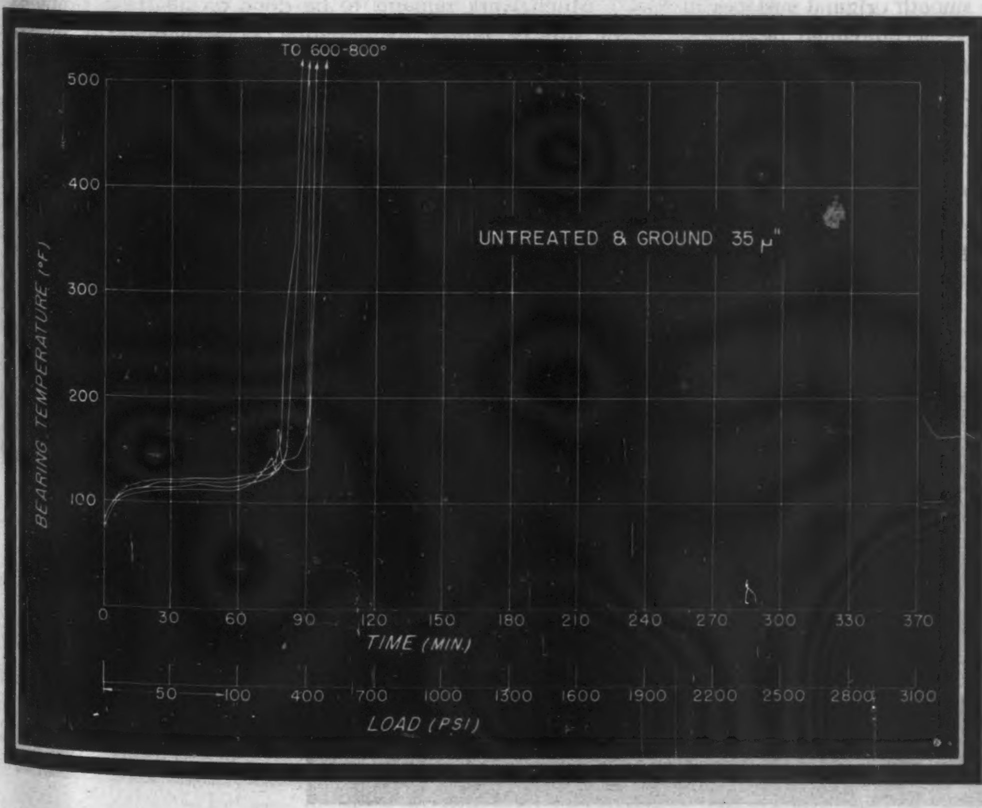


Fig. 7—Left—Test results on coarsely ground soft shafts. Average load was 400 pounds per square inch

caused little rise, but only one or two severe welds would cause a rise of 100 to 500 degrees.

Considerable work has been done in comparing the performance of hardened, ground and Superfinished shafts with shafts hardened and ground to a "good" grind of 10 microinches profilometer reading. These conditions were chosen because they represent a commercial possibility for improvement of the usual ground finish. A large number of tests indicate a breakdown load capacity for Superfinished shafts that is 60 to 70 per cent higher than for ground finishes. *Figs.* 4 and 5 show representative results of tests run on 4 shafts of each finish. Rise in temperature of the bearing is plotted against increase in time and loading. It is believed that moderate loading would show a still greater superiority of the Superfinished shafts, because grinder ridges would continually penetrate the oil film to some extent while a smooth surface would almost completely float on a film that was not overloaded. Action of the torque arm at medium pressures substantiate this idea because skipping was reduced.

The possibility that soft shafts, Superfinished then chromium plated, might be a practical application because of the fact that chromium has much less welding affinity for bearing metals than steel, had been in mind for some time. Consequently, a number of soft shafts were thus prepared. It was quickly found that "milky" chrome plating was unsatisfactory but that "bright" plating was highly successful. Thorough testing of bright plates .0002-in. thickness showed a breakdown load approximately half way between hard-Superfinished and hard-ground shafts. It is believed that a thicker plate of perhaps .0005-in. might be better than either because failure occurred through crushing of the plated case. *Fig.* 6 shows the results of tests with the 2-tenths plate.

An outstanding characteristic of the Superfinished hard shafts, and to an even greater extent of the chromed shafts, was a tendency for the smooth original surfaces to become still smoother. This was accompanied by an exceedingly low torque or coefficient of friction, and was the result of a very small weld-particle size. Advantages of chromium plating shaft journals instead of hardening them would be cheapness, shorter time for completion of the shaft, and elimination of straightening and subsequent partial return to crookedness during use. It would also be possible to use a treated high-tensile steel, unsuitable for carburizing, and still have a hard wear-resistant surface. This process is actually in large production.

Emphasis should be placed on the necessity for proper plating technique, and on the fact that chrome plating of either shaft or tool surfaces that are not very smooth is worse than useless. The current density, temperature, and concentration of the bath must be such that a bright plate results. Milky plate will not support 25 per cent of the load carried by the bright, and cannot be saved by polishing or Superfinishing. It is also necessary to remove hydrogen embrittlement by tempering at 300 to 400 degrees after plating.

Direction of Scratch Is Important

Widespread experience has shown that coarse surfaces heat, have high coefficients of friction and will not carry a reasonable load successfully as indicated in *Fig.* 7. There is never any possibility that they can be run-in to satisfactory operation. Yet the difference in the depth of scratches on a coarsely ground and a finely finished surface is measured in millionths of an inch. Very minute conditions make the difference between good and bad surfaces. Even the direction in which the scratch extends with reference to that of the shaft rotation or tool movement seems influential. Is it not reasonable to believe that a ridge running across the direction of motion would offer a more discontinuous path to break up the progress of damage than one running with the direction of motion as in grinding?

A limited amount of work by the author indicates that cross-direction, closely spaced, fine scratches do increase load capacity. Welds formed on such surfaces should be restricted in size, and should immediately be detached just like chips in an intermittent cut on an engine lathe. It has been determined that such surfaces form a glaze more rapidly than others of the same scratch depth.

Much work remains to be done on all these problems of surface pattern, lubrication, combinations of metals, and other methods of control of weld particle size. Yet much of the improvement that will be made in our machine tools, in the appliance field, in power-producing units, etc., must be based on greater accuracy in dimension and shape of the parts, as well as on finish.

It is a pleasure to acknowledge the efficient manner in which M. J. Brunk, research engineer, conducted the tests discussed in this article.

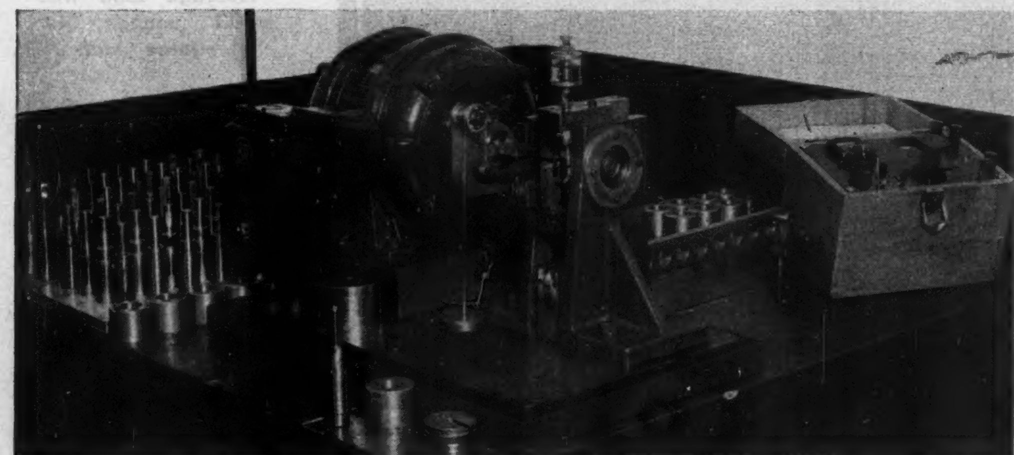
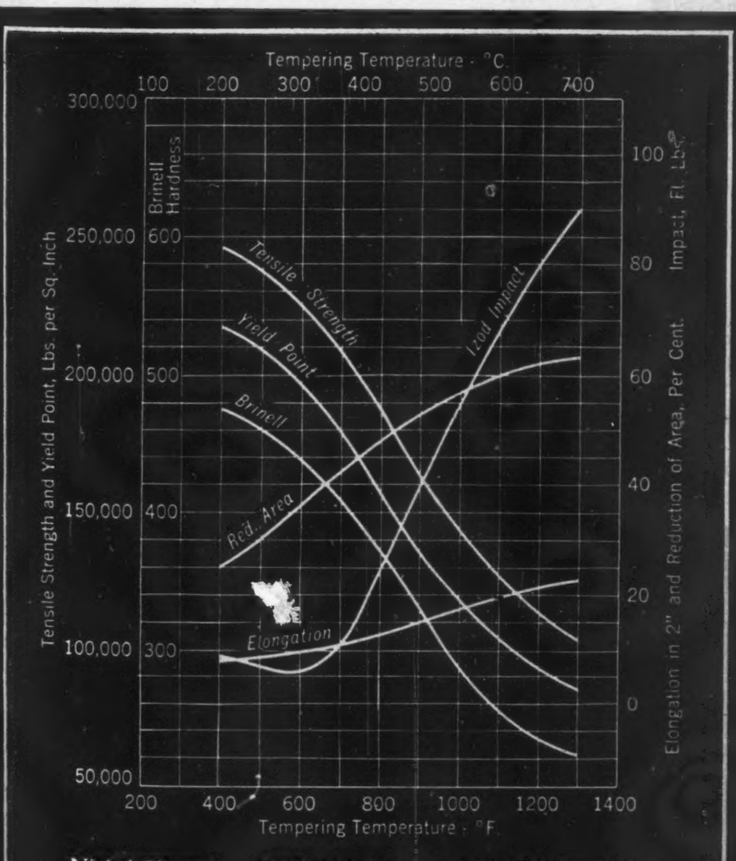


Fig. 8—Left—Machine used for making tests on bearings

Selecting Steel on the Basis of Hardenability

By A. L. Boegehold
Research Laboratories Division
General Motors Corporation

Part I—Interpreting Data from Curves



Nickel-Chromium Steel, S.A.E. No. 3140, Oil Quenched
(In small sizes, ½" to 1½" diameter or thickness)

Composition, %		*Approx. Critical Range*	*Heat Treatment*
Carbon	0.35-0.45	Ac_1 1355° F.	Quenched from 1475°-1525° F.
Manganese	0.60-0.90		
Phosphorus	0.04 Max.	Ac_2 1380° F.	Quenched into Oil
Sulphur	0.05 Max.		
Nickel	1.00-1.50		
Chromium	0.45-0.75	Ac_3 1395° F.	Tempered as Indicated

SELECTION of steel for some part of a machine seems to be a simple procedure when regarded superficially. One merely selects a steel that will harden properly, tempers it to the strength and hardness desired, and the job is done. However, there are so many different steels of varying hardenability and so many sizes and shapes of parts to be made that fairly precise rules are needed to govern the choice of steels. If one tries to set forth all the facts affecting the ultimate selection, it becomes apparent that this selection is in reality a complicated business.

Reasons for selecting a steel may be divided under two main classifications: (1) Those that relate to the economics of producing the part and (2) those that relate to suitability from an engineering standpoint. Under the first heading come:

a. Annealing characteristics
b. Machinability
c. Ease of manufacture at the mill
d. Ease of control within quality and composition limits
e. Relation to other steels in use
f. Cost.

Regardless of economic aspects, the steel selected must first of all be satisfactory from an engineering standpoint; that is, it must stand up in service. Whether it stands up in service is determined by the relation between the physical properties of the part and the stresses imposed on it. It is not necessary here to discuss this relationship in detail as it has been ably discussed at length in the literature by many authorities. Suffice it to say that if the engineer has a rough idea of what stresses will be imposed upon the part in service, he can decide what properties the part must have to resist those stresses successfully.

The next step is to pick a steel that can develop those desired properties in the part in question. As a final over-all check on

Abstract of a paper presented at a recent Metropolitan Section meeting of the Society of Automotive Engineers in New York.

Fig. 1 — Charts such as this become unnecessary when one learns how to interpret the overall data in Figs. 2, 3, 4 and 5

the validity of the various assumptions and decisions resulting in selection of a certain steel, the part is service tested. Then, and only then, can the success of the part be determined.

In selecting a steel on the basis of its potential physical properties it used to be customary to refer to physical property charts. All steel producers and alloy companies publish such physical property charts, of the type shown in *Fig.* 1, representing the effect of tempering temperature after hardening on tensile strength, yield point, reduction of area, per cent elongation and hardness. The data for these charts were obtained by testing tensile test specimens cut from one-inch diameter bars that had been

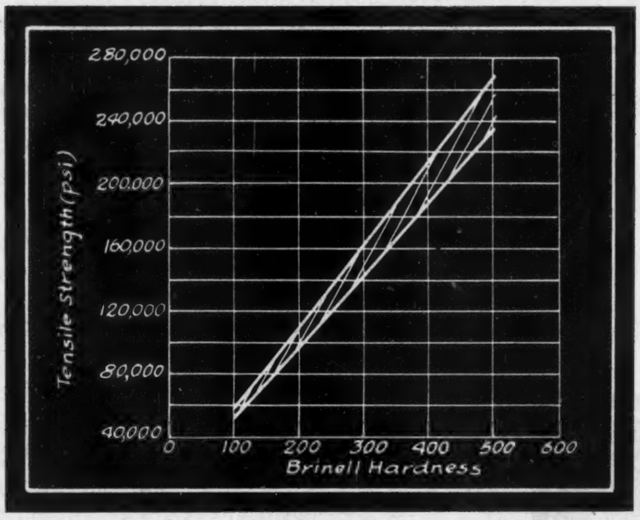
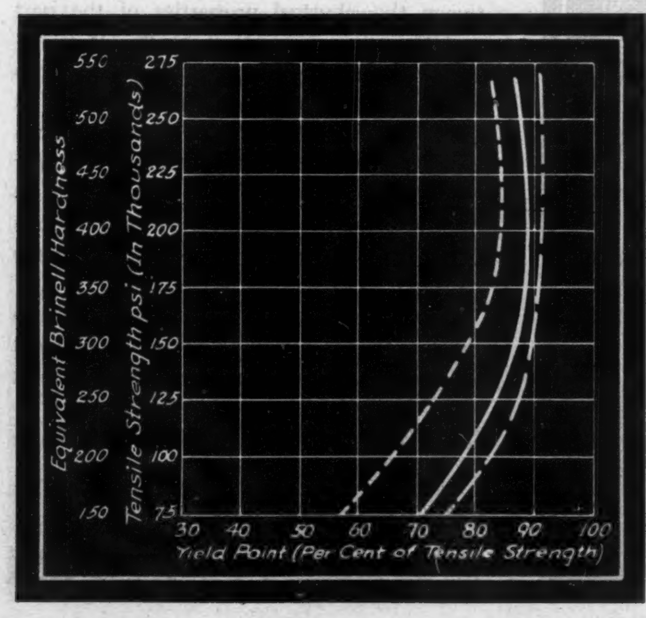


Fig. 2—Above—Relationship between hardness and tensile strength for carbon and alloy steels, series 1000, 1300, 3100, 3200, 4100, 2300, 4600, 5100, and 6100, in hardened and tempered, as-rolled, annealed and normalized conditions

Fig. 3—Below—Relationship between tensile and yield strength for all steels of series 1000, X-1300, 1300, 5100, 4100, 4300, 4600, 3100, 3200, 2300, and 6100, having .3 to .5 per cent carbon content, in the quenched and tempered condition. Solid curve shows location of most test points; broken lines define variations of remaining points. Tensile strength always is 500 times Brinell hardness



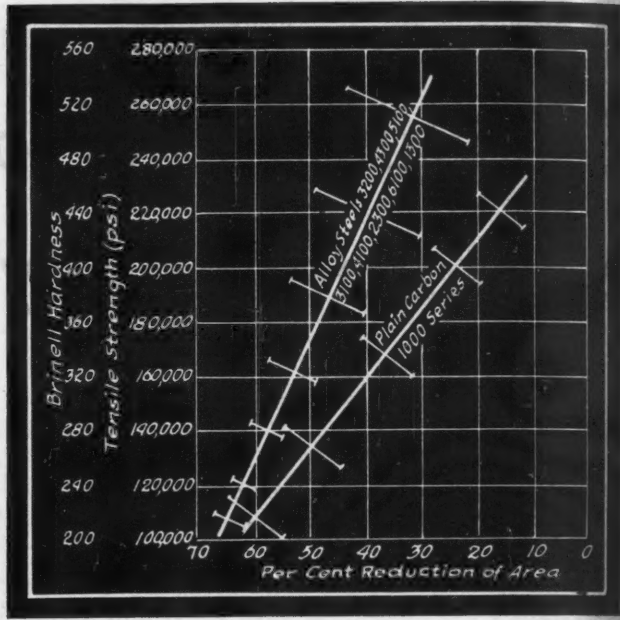


Fig. 4—This chart indicates relationship between tensile strength and reduction of area. Cross lines show variation from mean which may be caused by quality differences and magnitude of parasitic stresses induced by quenching

hardened and subjected to a variety of tempering treatments.

Some steels harden only part way through a one-inch diameter bar. The outside ¼-inch may harden fully but the center ½-inch may be only partially hardened. Therefore the physical properties in a test bar cut from this center ½-inch section would not represent the properties obtained in the ¼-inch outer layer which was properly hardened.

Physical property charts that have been based on data obtained on incompletely hardened test bars can be detected by the low hardness values shown for low tempering temperatures. For example, a .35 per cent carbon alloy steel should harden to at least 48 rockwell C to be fully hardened, so the hardness after tempering at 400 degrees Fahr. should be that high since 400 degrees Fahr. will not reduce the original hardness. When physical property charts for a .35 per cent carbon steel show 40 rockwell C or thereabouts after a 400-degree draw, as some do, it is evidence of incomplete hardening of the test bar and the properties shown in that chart are of doubtful value.

Correlation of Hardness and Strength

The use of inexact data of this kind, obtained without careful regard for the as-quenched hardness of test bars, was partly responsible for the supposition that each steel had a combination of physical properties peculiar to itself and different from other steels. Janitsky[1] showed that physical properties of different steels could be closely correlated with hardness and that the composition had little effect on this correlation. From this we have reasoned that it was not necessary to have physical property charts like *Fig.* 1 for every different type steel since one set of

[1] E. J. Janitsky and M. Baeyertz, A.S.M. *Metals Handbook*, 1939, Page 515.

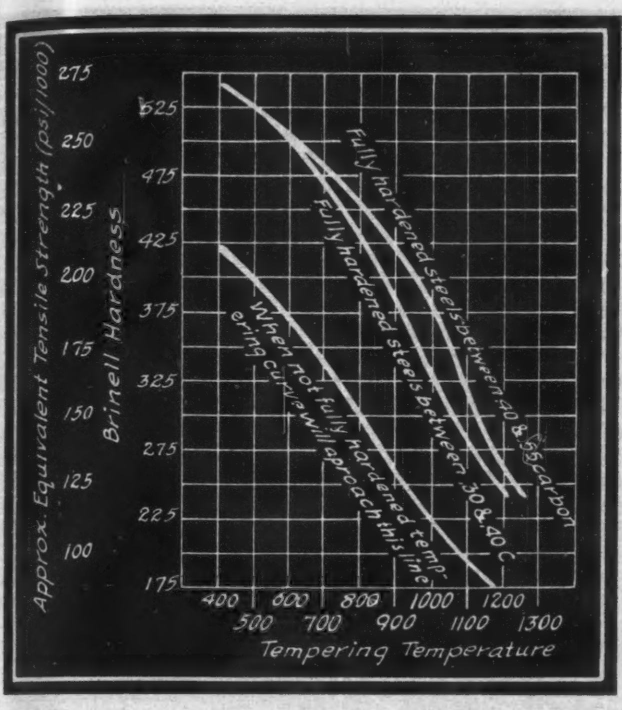


Fig. 5—Curves show effect of tempering on tensile strength of carbon and alloy steels including series 1000, 1300, 3100, 3200, 4100, 2300, 4600, 5100, and 6100

charts would show the relationship for all steels between hardness and—

1. Tensile strength
2. Yield point
3. Reduction of area
4. Tempering temperature.

The S.A.E. charts which indicate these properties are shown in *Figs.* 2, 3, 4 and 5.

Variation of properties within the bands shown in these charts is not the result of variation in composition. Properties of any composition steel may occur toward the upper or lower boundaries of the band, depending upon the quality of the steel which is a consequence of the details of procedure in making the steel; i.e., different heats of the same type of steel may show as much or more spread in the relation between hardness and physical properties than shown by heats of two different steels.

These few charts summarize the information presented by physical property charts for individual steels and perhaps emphasize the lack of information in these charts for selecting the right steel for each article to be made. We have to know, first, the maximum strength that a steel can develop and, secondly, how slowly the steel can be quenched and still obtain that maximum strength. Loads to be imposed on the piece determine what strength steel is needed, the size of the piece determines how fast it can be cooled by quenching, and this in turn determines how much alloy is needed to make the steel harden when cooled at the rate obtainable in the piece.

Maximum strength obtainable in steel is dependent on its carbon content. *Fig.* 6 shows how maximum obtainable hardness (and its corollary, strength) are affected by carbon content. This maximum hardness is obtainable in some steels only by means of a quench that is faster than those associated with section sizes ordinarily encountered in automotive practice. *Fig.* 6 tells nothing about how fast the steel has to be quenched to obtain the hardness indicated. As a better guide for deciding on hardness obtainable as related to carbon content, *Fig.* 7 shows a second curve in addition to the one in *Fig.* 6. The second curve is five to seven points rockwell C lower than the curve for maximum hardness obtainable and represents the lower limit of hardness obtained when the steel is quenched at 600 degrees Fahr. per second, a cooling speed commonly encountered in the processing of automotive parts, and also, as will be seen later, the cooling rate at the 1/16-inch point on the Jominy hardenability test bar. Hardenability curves for 1400 heats of steel collected by the joint S.A.E.-A.I.S.I. Hardenability Committee were studied to determine this curve. A third curve represents the lower limit for plain carbon steels at this same cooling rate.

This chart may be used to judge the amount of carbon necessary in a steel to provide the amount of hardness and, therefore, strength to meet the engineering re-

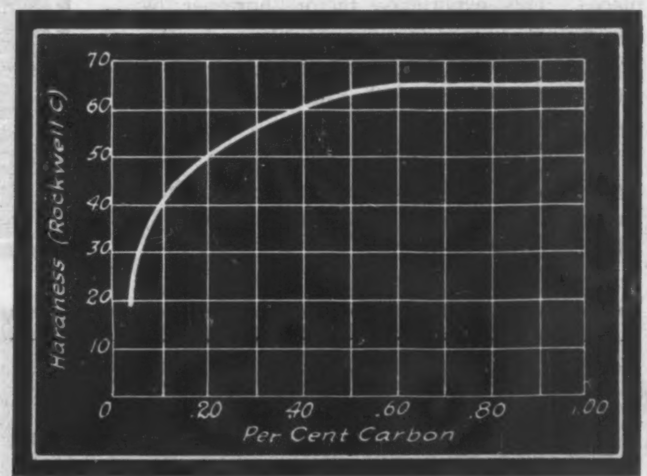


Fig. 6—Above—Maximum strength obtainable in steel is dependent on carbon content. This curve, prepared by Burns, Moore and Archer, shows the maximum hardness obtainable in plain carbon and low alloy steel

Fig. 7—Below—Solid line shows minimum hardness at 1/16-inch from quenched end of Jominy bar (cooling rate of 600 degrees Fahr. per second). These curves represent minimum values for 90 per cent of 1400 heats studied.

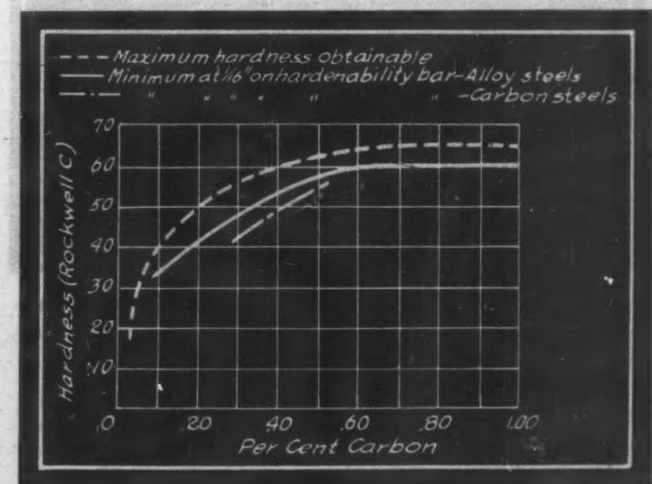
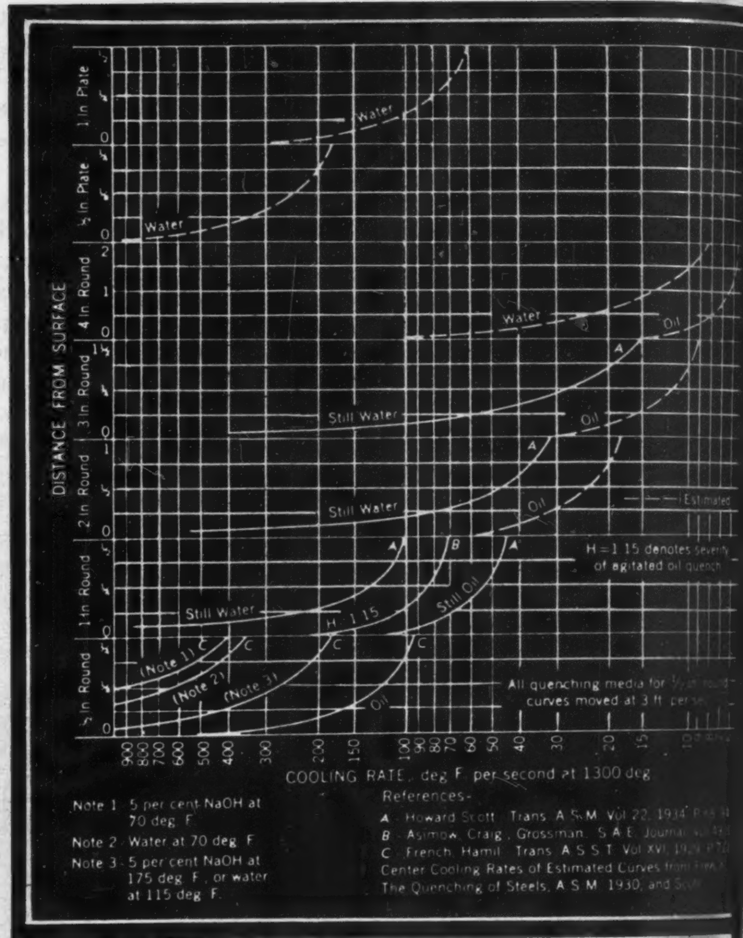


Fig. 8—Right—Cooling rate data from this chart can be correlated with Jominy end-quench test results when selecting steel



quirements of the piece or part under consideration. Due consideration must be given the reduction in hardness by tempering following hardening. This, however, is outside the scope of this discussion.

The next important consideration is the rates of cooling that will occur when the piece is being quenched for hardening. Certain steels may be capable of developing the required strength in light sections but not in heavy sections. As already mentioned, that cooling rate below which a steel fails to harden depends upon the alloy content and is the means by which hardenability of the steel is classified or graded.

Experienced metallurgists know from long practice about what kind of steel is required to obtain adequate hardening in a certain size piece. This experience factor, however, is now being reduced to an exact science by the process of collecting information on cooling rates in various size sections and hardenability data on all steels.

Most of the available information on cooling rates in different size sections when quenched in oil or water is shown in *Fig.* 8, which was prepared for the S. A. E. Hardenability Test Committee and now appears in the S. A. E. Standard book in the recommended procedure for hardenability testing. Another form of this information is shown in *Figs.* 9 and 10. This method of plotting permits more accurate determination of cooling rates in sizes between those shown in *Fig.* 8.

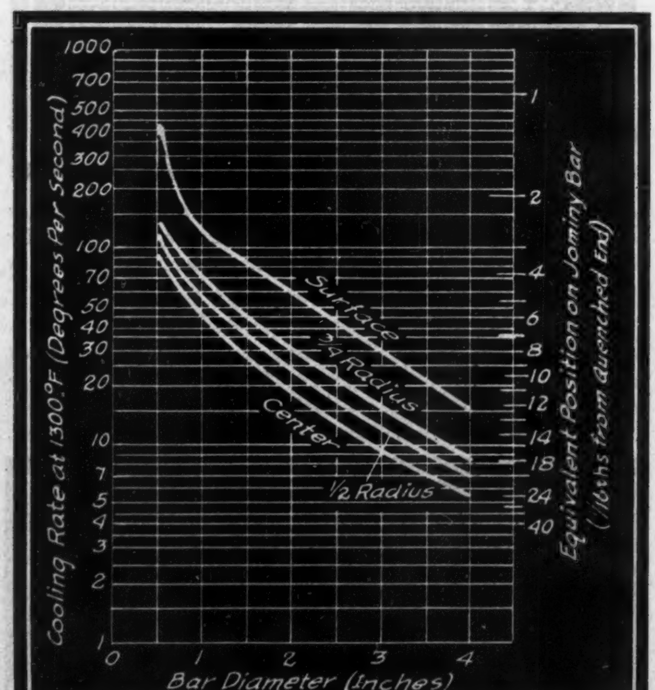


Fig. 9—Above—Cooling rates for all steels, quenched in still oil. Data is same as that in Fig. 8 except more accurate plotting of rates for all bar sizes is possible

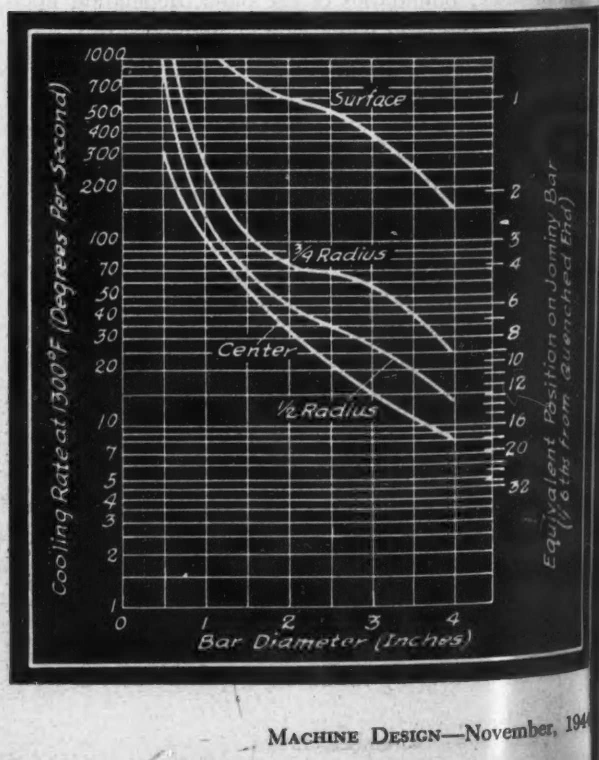


Fig. 10—Right—Cooling rates for all steels when quenched in still water. In this chart and that of Fig. 9, equivalent Jominy bar positions are correlated with rates

In these charts is depicted the manner in which cooling rates change with distance below the surface of the object quenched. The speed of cooling decreases rapidly from the surface to the center of a bar and is slower at corresponding locations as the bar size increases. The rate of cooling also differs, depending on whether oil or water is used as the quenching medium. We must know, therefore, how every steel reacts when subjected to each one of these cooling rates. This knowledge will permit us to predict how far below the surface of each size bar any steel will harden properly and also what hardnesses will occur at each cooling rate. The range of cooling rates ordinarily used varies from 10 degrees Fahr. per second to 1000 degrees Fahr. per second. Cooling rates at the surface of water-quenched rounds less than 2 inches in diameter are faster than 1000 degrees Fahr. per second measured at 1300 degrees Fahr. Cooling rates at the center of oil-quenched 3-inch rounds are as low as 10 degrees Fahr. per second.

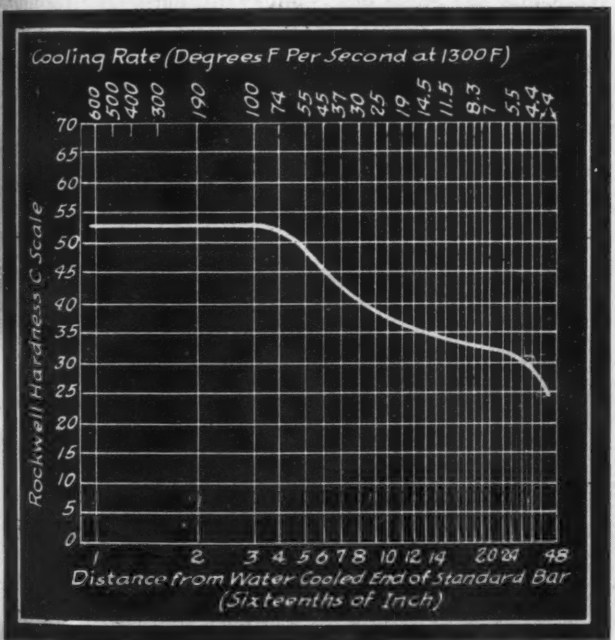


Fig. 11—Typical hardenability curve shows relationship between hardness and cooling rate for 5130 steel

To study the hardness produced in steels by quenching at rates of cooling within this wide range, a Jominy hardenability test is made[2], this being conducted on a cylinder of steel three inches long and one inch in diameter. The cylinder is heated to the proper temperature for hardening and then is quenched on one end only by a stream of water. This method of quenching produces cooling rates varying from 600 degrees Fahr. per second at 1/16-inch from the quenched end down to 4 degrees Fahr. per second at the upper end. We do not need to study cooling rates faster than 600 degrees Fahr. per second because at that speed full hardness is obtained with practically all steels currently used. We next make hardness determinations at intervals of 1/16-inch all along the bar and plot a curve of hardness versus distance from the quenched end. Then, in place of distance from the quenched end we substitute cooling rates occurring at each 1/16-inch. This gives us the desired connection between hardness and cooling rate.

A typical hardenability curve is shown in *Fig.* 11. From this curve one may predict what hardness can be obtained at the surface and between the surface and center of any size round, quenched either in water or oil. It becomes a simple matter of substituting hardness values from the hardness-cooling-rate curve in place of cooling rates in the curves shown in *Figs.* 8, 9 and 10.

If one is interested in selecting a steel for a 2-inch round shaft to be hardened by quenching in oil, he refers to *Figs.* 8 or 9 and finds that the cooling rate at the center is 18 degrees per second, or the same as at about 11/16-inch on the Jominy bar. At half the radius below the surface the cooling rate is 24 degrees per second or at about 9/16-inch on the Jominy bar. At one-quarter the radius below the surface the cooling rate is 60 degrees per second or at 4½-sixteenths on the Jominy bar. The

TABLE I

Cooling Rates and Hardnesses—Oil-Quenched

Location in 2-in. diam bar	Cooling Rate in Oil-Quenched 2-in. diam bar (deg F per sec)	Corresponding Locations on Jominy Bar	Hardness for 5130 from Curve in Fig. 11 (Rockwell C)
Center	18	11/16	36
½ radius below surface	24	9/16	38
¼ radius below surface	32	8/16	41
Surface	60	4½/16	50

hardnesses at these points on the curve shown in *Fig.* 11 may be substituted to show that a two-inch round of 5130 steel quenched in oil from 1675 degrees Fahr. would have hardnesses throughout the cross section as shown in TABLE I. If the same bar were water-quenched the cor-

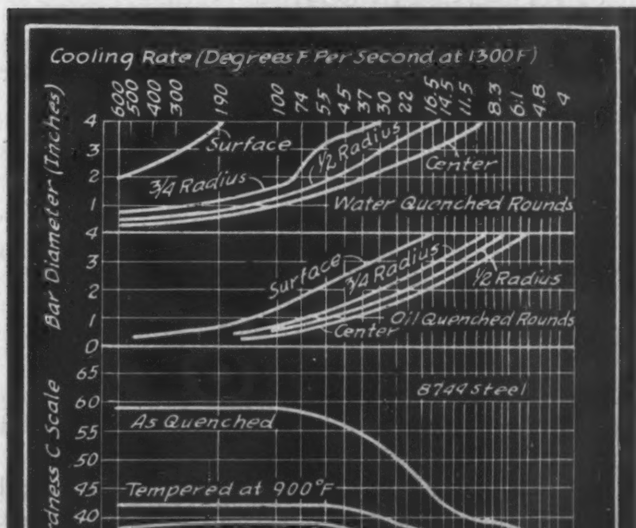


Fig. 12—Combining hardenability curve (bottom) with cooling rate curves (top) provides quick means for determining hardness throughout various size rounds

[2] Jominy and Boegehold, A.S.M. *Transactions*, 1938, Vol. 26, Page 574. S.A.E. *Standards*, Page 314.

responding cooling rates and hardnesses would be as shown in Table II.

Whether either one of these hardness patterns is considered suitable depends on what kind of loads are to be imposed and on the views of the person making the decision. Some metallurgists believe that for best physical properties after tempering, with 35 to 40 rockwell C giving a hardness range suitable for an axle shaft, the steel must first be hardened above 49 rockwell C at the surface and above 44 rockwell C at the center. Others accept any as-quenched hardness at the center so long as the center comes within the hardness limits required after tempering. Center hardness then could be as low as 39

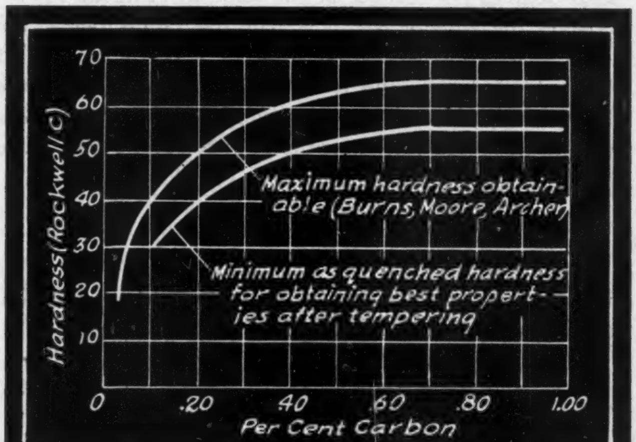


Fig. 13—Curve drawn below maximum hardness curve indicates minimum hardness which will temper to adequate physicals for majority of applications

to 40 rockwell C to stay within the 35 to 40 range after tempering. It is not proposed to argue this point in this paper because the facts are not available to prove one side or the other. For example, we have no fatigue test

Table II

Cooling Rates and Hardnesses—Water-Quenched

Location in 2-in. diam bar	Cooling Rate in Water-Quenched 2-in. diam bar (deg F per sec)	Corresponding Locations on Jominy Bar	Hardness for 5130 from Curve in Fig. 11 (Rockwell C)
Center	27	7½/16	39
½ radius below surface	46	6/16	45
¼ radius below surface	70	4/16	52
Surface	600	1/16	53

results on partially hardened and tempered steel to compare with fatigue tests on fully hardened and tempered steel of the same hardness and composition.

If one belongs to the school that is satisfied with partial hardening throughout the section and wants a hardness after tempering of 35 to 40 rockwell C, he would consider 5130 steel satisfactory for the part. By tempering at about 900 degrees Fahr., the part of the bar that hardened to around 50 rockwell C would be softened to 40 rockwell C and the center that was hardened to 39 rockwell C would temper to about 35 rockwell C and the whole bar section would be within the desired hardness limits. The greater hardness decrease obtained during tempering fully hardened steel than occurs in partially hardened steel is shown in *Fig.* 12, giving the as-quenched hardenability curve for 8744 steel and the hardness curves obtained by tempering at 900 degrees Fahr. and 1000 degrees Fahr. That part of the bar that hardened to 59 rockwell C, when tempered at 900 degrees Fahr., softened to 42 rockwell C, a drop of 17 points. When the quenched hardness was 50 rockwell C, 900 degrees Fahr. tempering only reduced the hardness 11 points to 39 rockwell C. When the quenched hardness was 40 rockwell C, 900 degrees Fahr. only reduced the hardness 4 points to 36 rockwell C. *Fig.* 12 also illustrates how the chart of cooling rates in various diameter bars may be joined to the hardenability curve chart for the purpose of using hardenability curve or tempered hardenability curve for predicting hardnesses that will be obtained in any size bar with an oil or water quench.

Securing "Full Hardening"

Metallurgists who require minimum 44 rockwell C at the center and those who require full hardening throughout the section, or at least to three-quarter-radius below the surface would reject 5130 steel for this part. When we refer to "full hardening" we are again talking in terms on which all metallurgists are not on common ground.

Each steel, depending upon its carbon content, is capable of being hardened to a certain maximum hard-

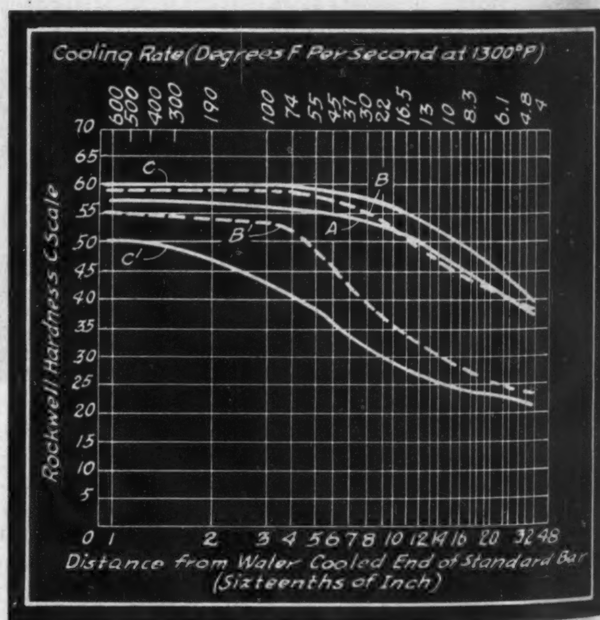


Fig. 14—Curve A is for steel that will harden to 50 rockwell C at center of 2-inch, oil-quenched round. B-B' show expected limits of 94 per cent of heats of 8740 and 8640. C-C' show extreme limits for same steels in 168 heats

ness. This maximum hardness determined by carbon content is depicted in *Fig.* 6. The quenching speed required to obtain this maximum hardness is faster than ordinarily encountered in the size sections common to automotive practice, so we have to accept a somewhat lower hardness. For example, a .40 per cent carbon steel is capable of hardening to 60 rockwell C when quenched fast enough to obtain 100 per cent martensite. The speed of quench

(Continued on Page 166)

Novel Eccentric Linkage Permits Extended Dwell

By James J. Kux
Kux Machine Co.

TO MEET combined requirements of speed, high pressure and extended dwell, a special mechanism has been developed to perform this task on an automatic tablet press for compressing powder metals. Consisting of an eccentric strap capable of withstanding the high pressure, the mechanism is jointed in such a way that a cam action governs the movement of this joint to allow a 60-degree dwell during the operating cycle as shown in *Fig.* 1.

To indicate the importance of this unusual mechanical motion a brief description of the press cycle follows: A feed shoe containing the powdered material deposits a charge into a die cavity mounted in a die table at the front of the machine, *Figs.* 2 and 3. Forming the bottom of the die is a lower punch the height of which in the die cavity, controlled by a simple screw adjustment, determines the amount of material deposited in the cavity. An upper punch travels downward, entering the die partially to compress the material while the lower punch travels upward, applying its compressing pressure to the bottom of the material which accordingly becomes compacted into a solid tablet. The upper punch then travels upward out of the cavity, while the lower punch moves further upward ejecting the compacted tablet from the die. There it is pushed off the die table by the action of the feed shoe, returning to refill the die cavity with another charge. The lower punch, after ejecting the tablet, returns to its down position to determine again the amount of refill.

Upper Punch Dwell Needed in Cycle

A dwell during the period pressure is being applied by the upper punch is essential for the manufacture of large shaped parts, such as a flange-type bushing. To form this part, it is necessary that the upper punch enter the die cavity, compressing the loose powder which forms the flange, and then dwell while the lower punch rises to compress the lower section of the bushing. Since the stroke of the upper punch in the die cavity is comparatively short, while the lower punch pressure stroke is much longer, the dwell imparted to the upper punch is necessary to permit the lower punch to attain its full stroke.

Normally a dwell or pause is obtained by a cam operating a bell-crank lever which in turn operates the punch-carrying mechanism. This design, however, is only suit-

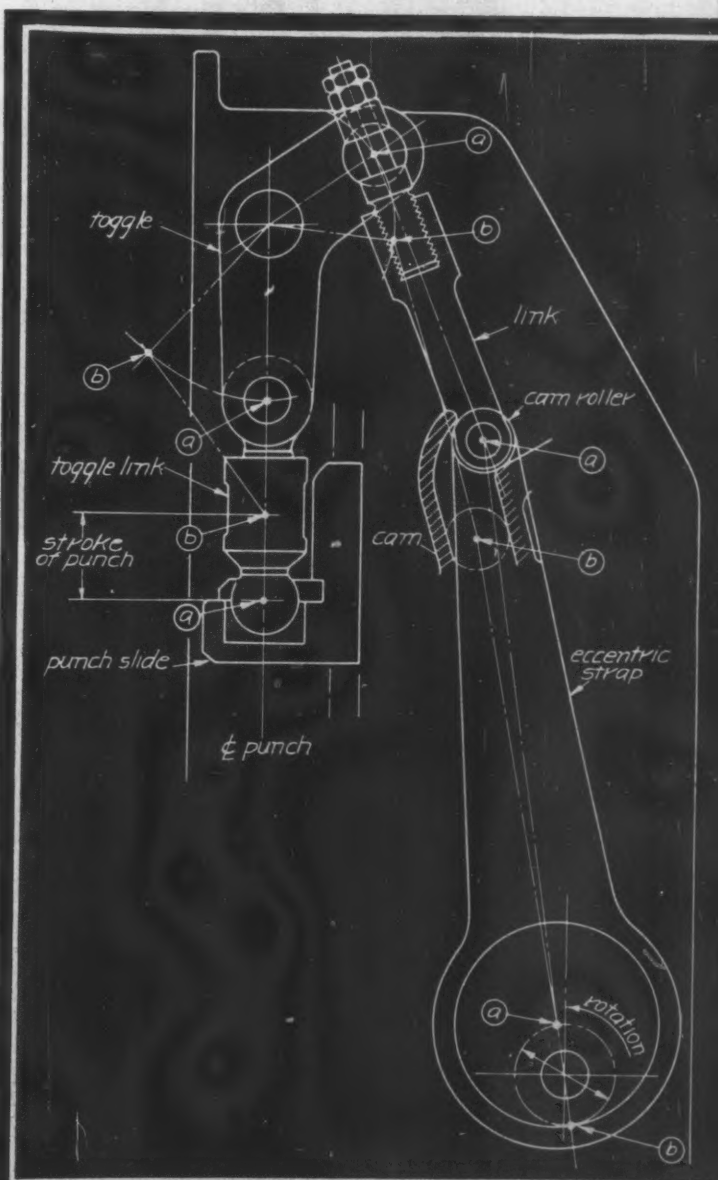


Fig. 1—Detail of action of cam-controlled eccentric strap to produce a 60-degree dwell at high pressure

able for applying total compressing pressures under about 50 tons. A cam face contacting a follower roller is a poor pressure-applying mechanism because of the lack of surface area contact between the cam face and follower roller, there being only a line contact at any time. When properly designed for the stresses and strains involved in applying over 50 tons pressure, a cam and roller becomes so large that its use is not feasible or practical.

For this press which develops 150-ton pressure it was therefore necessary to turn to a mechanism of a design capable of functioning under this high pressure. An eccentric strap operated by an eccentric cam with ample

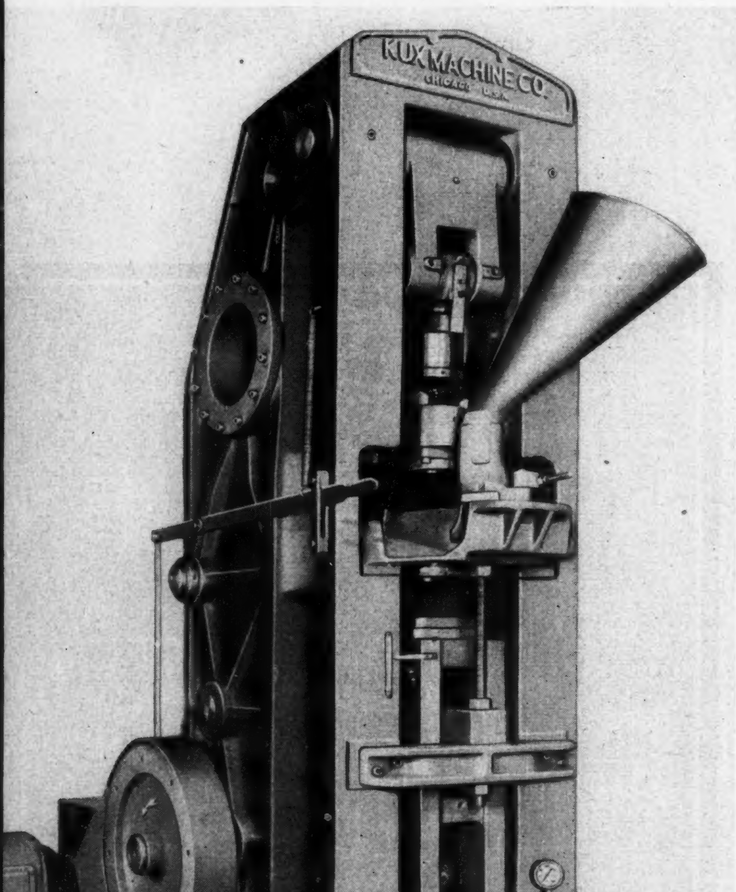


Fig. 2—Combined mechanical-hydraulic press develops 150-ton pressure at ten cycles per minute

bearing surface between the two for withstanding the stresses and strains involved was chosen, *Figs.* 1 and 3.

This eccentric strap is connected to a strong toggle linkage for operation of the upper punch. To obtain the necessary dwell, using this eccentric cam, the eccentric strap is jointed at a point about two-thirds of its length. The pivoting pin of this joint is extended beyond the width of the strap and a cam follower roller is mounted thereon. A properly designed groove cam is mounted to the inside of the side frame in line with the joint of the eccentric strap so that the follower roller turning on the joint pin of the strap operates in the groove of the cam.

Thus as the lower main section of the eccentric strap moves upward through the full throw or stroke of its eccentric cam, the upper secondary section of the strap partially moves upward and partially moves horizontally through an arc created by the curve of the groove cam which directs the follower roller on the eccentric strap joint.

Because of this cam action, the last 30 degrees upward movement and the first 30 degrees downward movement of the main section of the strap produce no upward or downward motion at the top end of the secondary section of the strap. In other words, the main section moves upward 180 degrees of its cycle and downward 180 degrees since it is operated by its eccentric cam but the secondary section of the strap moves upward during 150 degrees, pauses 60 degrees and then moves down during 150 degrees because of the action of the groove cam directing the joint of the strap through an arc.

Compact Design Withstands High Pressure

Since the top end of the secondary section of the strap is pivotly connected to the toggle linkage operating the upper punch, all of this mechanism stays on dwell or pause for this full 60 degrees. The toggle mechanism is timed so that the dwell takes place while the toggle is closed on dead center in its pressure position and while the upper punch mounted to the toggle slide is applying pressure to the loose powder in the die.

Besides producing the necessary upper punch pressure dwell, this design permits use of parts which will safely withstand high pressures without becoming so large that their size precludes their use in this type of machine. The full compressing pressure is applied through the hardened and ground alloy steel eccentric cam and its bronze-bushed strap with ample bearing area between the two, to a large hardened alloy steel pin at the eccentric strap joint and another similar pin connecting to the toggle linkage. These pins are also amply strong with sufficient bearing area. The toggle linkage is composed of hardened alloy steel pins, links of steel castings and hardened and ground ball joint connecting to the punch-carrying slide.

Hydraulic Operation Utilized for Lower Punch

Although the upper punch is mechanically operated, the lower punch pressure mechanism is hydraulically operated. This combination of mechanical-hydraulic mechanism has the distinct advantage of providing the high speed automatic operation of a mechanical press along with the uniformly applied high pressure capacity of a hydraulic press.

A self-contained hydraulic unit complete with necessary pump, valves, oil storage tank, piping, etc. as illustrated in Fig. 4, is mounted to and within the main frame of the machine. To provide the hydraulic ram with sufficient speed in rising, a nitrogen accumulator bottle is incorporated into the hydraulic system. Use of this accum-

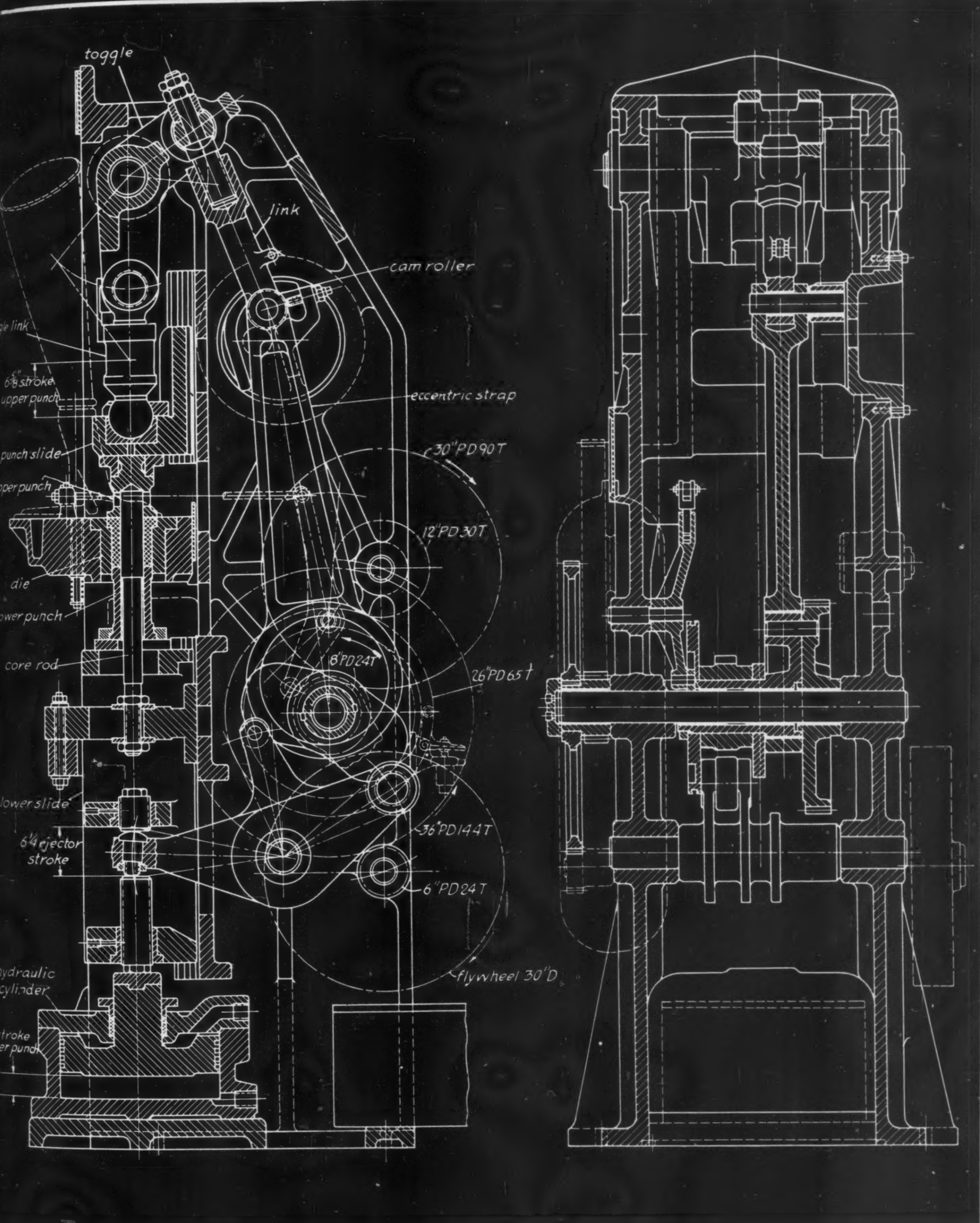


Fig. 3—Sectional view of press showing combined mechanical and hydraulic actions on tableting dies. Cam-controlled eccentric linkage allows 60-degree dwell to permit hydraulic ram on lower punch to exert 150-ton squeeze

lator enables the hydraulic ram in exerting its 150-ton pressure to rise and return as much as 3 inches as many as 8 to 10 times a minute.

Oil is delivered from a vane-type pump at up to 1000 pounds per square inch pressure to a master four-way valve and to the accumulator bottle. Interconnected in the line is a relief valve for adjusting the pressure of the oil delivered from the pump, regulating the ultimate pressure applied by the hydraulic ram.

Pilot Valve Is Cam Operated

The master four-way valve which directs the flow of oil to either the lower chamber of the hydraulic ram during the pressure period or to the upper chamber of the ram during its return is controlled by a pilot valve. The pilot valve is operated by a cam which is an integral part of the mechanical mechanism and is set to depress the valve at the correct time during the cycle of the machine.

When the full pressure of the machine is being utilized, the accumulator bottle is charged with gas under 500 pounds per square inch pressure. Oil from the hydraulic pump compresses the gas to 1000 pounds per square inch pressure. At the proper time in the cycle of the machine, the four-way valve opens its port to the oil line connected to the lower chamber of the hydraulic ram. Oil both from the pump and from the accumulator bottle flows at high speed, due to the action of the expanding gas in the bottle, to raise the ram at full pressure. The ram being 20 inches in diameter has 314 square inches area thus with 1000 pounds per square inch oil pressure being delivered to it, the ram has a total effective pressure of 314,000 pounds or 157 tons.

As soon as full stroke of the ram has been attained, the four-way valve is automatically reversed by its pilot valve and the flow of oil is directed to the upper chamber of the ram for its return stroke. The lower line is also opened to allow oil below the ram to flow back to the tank.

Interconnected in the line between the master valve and the upper chamber of the ram is a pressure-reducing valve which reduces the oil pressure used for returning the ram. While the ram is being returned and during the mechanical portion of the cycle when the formed tablet is being ejected, the die cavity is being refilled and the upper punch is traveling back downwards to compress the material partially, oil is again delivered to the accumulator bottle, recompressing its charge of gas.

The timing of the machine is such that the hydraulic ram rises to apply its pressure while the upper toggle is closed on center during its 60-degree pause period. Thus the full pressure applied to the tablet being compressed is absorbed by the toggle mechanism which is sufficiently strong for this duty.

The ejection stroke of the lower punch, when the finished tablet is being pushed up and out of the die cavity, is accomplished mechanically. Inasmuch as this action does not require as much pressure as does forming the tablet, a bell crank operated directly from a separate cam is used to raise the lower punch slide at the proper time. Adjustment for stroke is simply made by turning a control screw in the punch slide.

Toggle-Link Adjustment for Pressure

Pressure adjustment of the upper punch, for determining the depth of stroke of the punch in the die cavity, is easily made by a screw adjustment which shortens or lengthens the toggle link having its seat in the ball joint of the upper punch carrying slide. Pressure adjustment of the lower punch is made at the previously mentioned relief valve incorporated into the hydraulic system.

All of the mechanical mechanism of the machine is housed and guarded inside of the main frame. Steel castings are used for the frame and all working parts subjected to stress. The die table is supported by an extension of the frame, thus all pressures applied within the die are partially absorbed by the main frame itself.

The flange-type die is fully supported by a counterbore in the die table and secured down by socket screws, thus is easily inserted or removed. Both upper and lower punches, held by flange-type punch holders, are on slides which operate in adjustable V-gibs. All core rods used for parts requiring cored holes are fastened to a stationary core bracket mounted to the main frame directly below the lower punch.

With this simple tool mounting design, a set of punches, die and core rod can be inserted or removed from the machine in very little time.

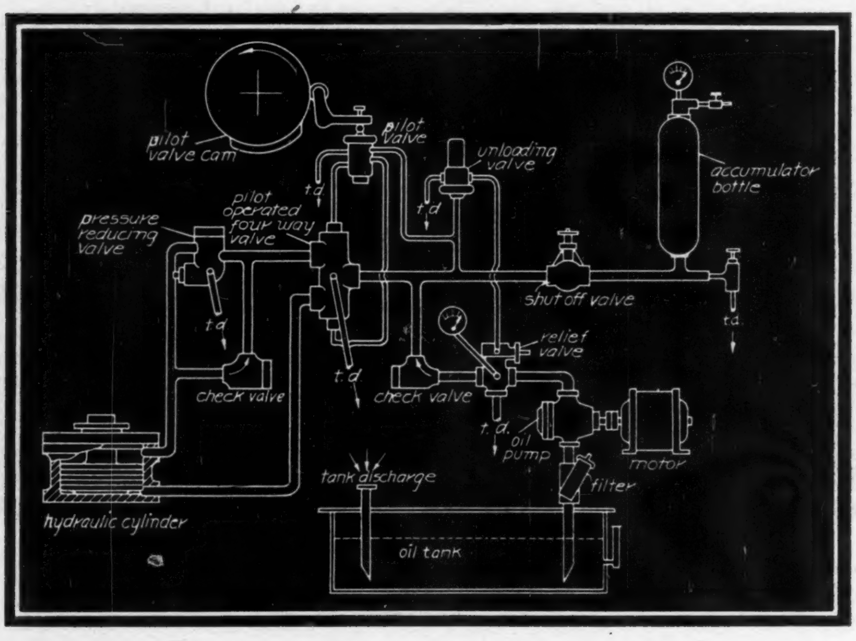
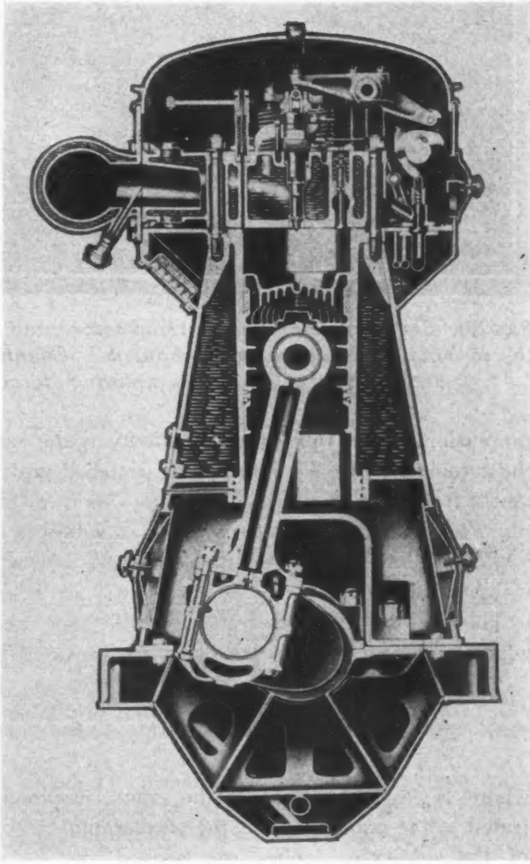


Fig. 4—Left—Simplified hydraulic circuit diagram for control of ram for operation of the lower die

Vibration and Noise—Causes and Cures

By Colin Carmichael

Part IV—Unbalance



Photo, courtesy Joshua Hendy Iron Works

Fig. 36 — Above — Example of a machine in which unbalanced forces and moments are an important factor is this marine diesel

VIBRATION mountings and isolating materials, discussed in Parts II and III of this series, eliminate only a certain percentage of the vibration and noise emanating from a machine or mechanism. For maximum effectiveness, therefore, it is sound practice first to reduce as far as practicable the origin of the vibration. Chief sources of trouble are moving parts with reciprocating or oscillating motions, and it is the purpose of this article to review the principles used in determining and correcting the unbalanced forces due to such parts, also to present a general method of solution applicable to any type of moving part.

Inasmuch as the periodic force due to a reciprocating or oscillating part is equal to the mass of the part times its acceleration, analysis of the acceleration is the first step in determining the magnitude of the force.

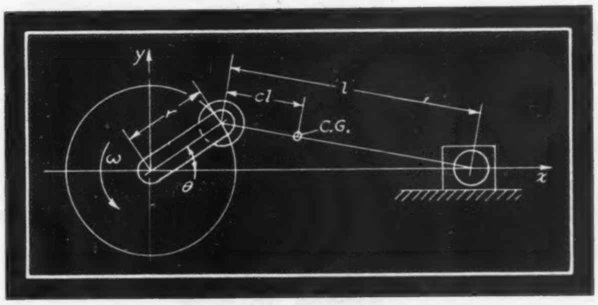


Fig. 37 — Left — First step in analyzing mechanism for unbalance is to establish leading dimensions. For force analysis, connecting rod mass is assumed concentrated at pivot points

For the simpler types of mechanisms it is possible to analyze the motion mathematically, so that once the equations of motion have been determined the forces can be found by simple substitution. When the motion is complex the mathematics also become complex and it often is simpler to employ a graphical solution which, however, is applicable only to a mechanism having the particular dimensions employed in the analysis.

Typical of the mathematical approach is the method followed in dealing with the slider-crank mechanism which is so widely used for engines, compressors, pumps, presses, etc., *Fig*. 36. Derivations may be found in textbooks (1)*, hence only the final results and their significance will be discussed here. The method of using these results is perfectly general and may be applied to the other mechanisms.

Of the three moving parts, the crank generally has uniform rotation, while the slider has rectilinear (straight-line) motion. The connecting rod has a complex motion, one end rotating with the crank and the other moving with the slider; the resulting oscillation of the rod causes a rotating couple in addition to the forces caused by the translation motion. It can be shown that the forces due to linear motion of the rod are the same as those that would be caused by a rod of "dumbell" shape having all the mass concentrated in two lumps at the crankpin and wristpin respectively, each mass being so

*References in parenthesis are listed at end of article.

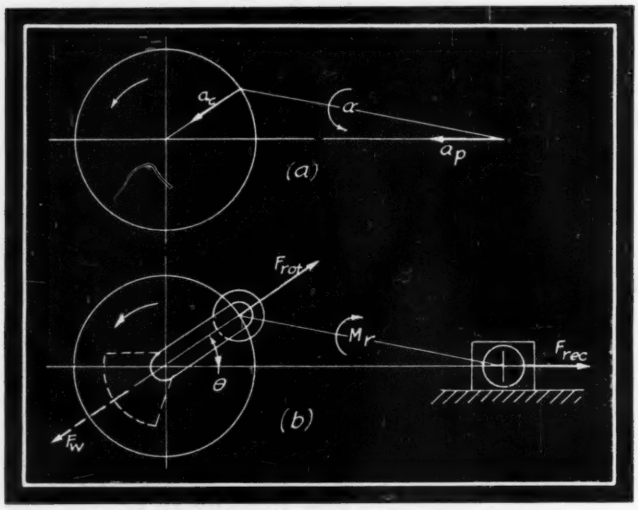


Fig. 38—Shows accelerations (a) and forces and moments (b) in simple slider-crank mechanism. Counterweight, shown dotted, balances rotating force

proportioned that the center of gravity is the same as in the actual rod, *Fig.* 37. Thus, if the center of gravity is distant a fraction c of the length of the rod from the crankpin, *Fig.* 37, the weight at the crankpin would be $(1\text{-}c)W_r$ while that at the wristpin would be cW_r, W_r being the weight of the rod.

The crank itself together with that part of the rod which is considered to rotate with it causes a force

$$F_{rot} = -\frac{W_c + (1-c)W_r}{g} a_c \qquad (1)$$

where W_c is the weight of the crank, assumed concentrated at the crankpin, a_c is the acceleration of the crankpin, inches per second per second, and g is the acceleration due to gravity, 386 inches per second per second. Likewise the piston itself (W_p) and that part of the rod which moves with it causes a force

$$F_{rec} = -\frac{W_p + cW_r}{g} a_p \qquad (2)$$

and the connecting rod causes a couple or moment, inch-pounds

$$M_r = -\frac{W_r \rho^2}{g} \alpha \qquad (3)$$

where ρ is the radius of gyration of the rod about its center of gravity, inches, and α is the rod's angular acceleration, radians per second per second.

For the analysis of vibrating forces it therefore is necessary to determine the three accelerations indicated on *Fig.* 38*a*. The corresponding forces are shown on *Fig.* 38*b*. The force vectors F_{rec} and F_{rot} are drawn to a length proportional to the force, the arrow pointing in the direction of the force exerted by the moving parts on the frame of the machine at the instant.

For the crank the normal acceleration, inches per second per second, is

$$a_c = -\omega^2 r \qquad (4)$$

where r = crank radius, inches, and ω = angular velocity of crank, radians per second = $(2\pi/60)$ times the revolutions per minute.

For the piston or slider the acceleration, inches per second per second is

$$a_p = -\omega^2 r[A_1 \cos\theta + A_2 \cos 2\theta - A_4 \cos 4\theta + A_6 \cos 6\theta - etc.] \qquad (5)$$

where $A_1 = 1$

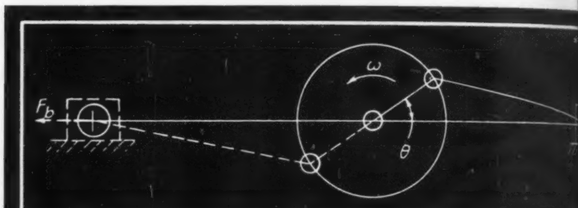
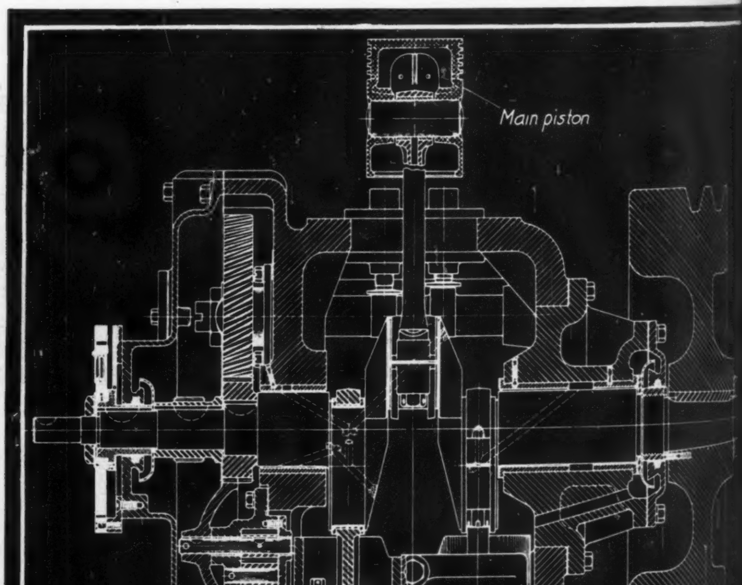


Fig. 39—Above—Complete counterbalance of reciprocating force may be effected by providing duplicate mechanism with opposite acceleration

Fig. 40 — Below — Single-cylinder high-speed fuel testing engine employs "bob-weights" in crankcase to balance piston, as shown in Fig. 39

Drawing, courtesy Waukesha Motor Co.



$$A_2=\frac{1}{n}+\frac{1}{4n^3}+\frac{15}{128n^5}+etc.$$

$$A_4=\frac{1}{4n^3}+\frac{3}{16n^5}+etc.$$

$$A_6=\frac{9}{128n^5}+etc.$$

In the foregoing, n = ratio of connecting rod length to crank radius (*Fig.* 37) $=l/r$.

Table I

Calculation of Primary Harmonic

Shaft angle θ	Ordinate	Value of y	$\cos\theta$	$y\cos\theta$	$\sin\theta$	$y\sin\theta$
0	y_0	1333	1.0000	1333	0	0
15	y_1	1284	.9659	1239	.2588	333
30	y_2	1088	.8660	944	.5000	544
45	y_3	786	.7071	555	.7071	555
60	y_4	429	.5000	215	.8660	371
75	y_5	77	.2588	20	.9659	74
90	y_6	−222	0	0	1.0000	−222
105	y_7	−441	−.2588	114	.9659	−425
120	y_8	−571	−.5000	285	.8660	−494
135	y_9	−628	−.7071	443	.7071	−443
150	y_{10}	−644	−.8660	557	.5000	−322
165	y_{11}	−648	−.9659	626	.2588	−168
180	y_{12}	−667	−1.0000	667	0	0
195	y_{13}	−706	−.9659	681	−.2588	183
210	y_{14}	−754	−.8660	653	−.5000	377
225	y_{15}	−786	−.7071	555	−.7071	555
240	y_{16}	−763	−.5000	386	−.8660	661
255	y_{17}	−655	−.2588	170	−.9659	633
270	y_{18}	−444	0	0	−1.000	444
285	y_{19}	−137	.2588	−35	−.9659	132
300	y_{20}	237	.5000	119	−.8660	−205
315	y_{21}	628	.7071	443	−.7071	−443
330	y_{22}	978	.8660	847	−.5000	−489
345	y_{23}	1227	.9659	1181	−.2588	−318
				Sum = 11998		Sum = 1333

$$A_1=\frac{11998}{12}=1000 \qquad B_1=\frac{1333}{12}=111$$

Resultant $=\sqrt{A_1^2+B_1^2}=1006$

Phase angle $=tan^{-1}(B_1/A_1)=tan^{-1}(111/1000)=6°-20'$

For the connecting rod the angular acceleration, radians per second per second, is

$$\alpha=\omega^2[B_1\sin\theta-B_3\sin 3\theta+B_5\sin 5\theta-etc.] \quad\ldots\ldots(6)$$

where $$B_1=\frac{1}{n}+\frac{1}{8n^3}+\frac{3}{64n^5}+etc.$$

$$B_3=\frac{3}{8n^3}+\frac{27}{128n^5}+etc.$$

$$B_5=\frac{15}{128n^5}+etc.$$

Substitution for a_c in Equation 1 gives

$$F_{rot}=\frac{W_c+(1-c)W_r}{g}\omega^2 r \quad\ldots\ldots(7)$$

This rotating force can, of course, be completely counterbalanced by a revolving weight attached to the crank, as indicated by broken lines in *Fig.* 38*b*.

Substitution for a_p in Equation 2 gives

$$F_{rec}=\frac{W_p+cW_r}{g}\omega^2 r\times$$

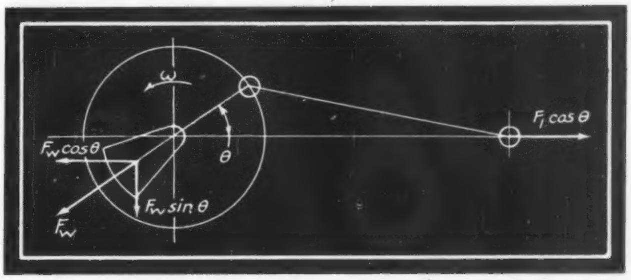
$$[\cos\theta+A_2\cos 2\theta-A_4\cos 4\theta+A_6\cos 6\theta-etc.]$$

$$=F_1\cos\theta+F_2\cos 2\theta-F_4\cos 4\theta+F_6\cos 6\theta-etc.\ldots\ldots(8)$$

First term on the right of Equation 8 is called the primary force, the second is called the secondary while the others are known as the fourth order, sixth order, etc. In this form of solution it is at once evident what the frequencies and magnitudes of the vibrating forces are, a factor of great significance in choosing vibration mountings.

Inasmuch as the connecting rod/crank ratio usually is not less than 3, the fourth, sixth and higher order terms (Equation 5) are of negligible importance except in multicylinder engines where they may add up to an appreciable amount.

How can the magnitude of the reciprocating force be diminished? Complete balance can be secured only by



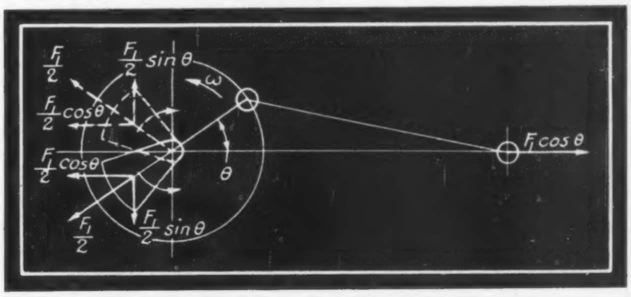


Fig. 41 — Above — Revolving counterweight can only partially balance primary reciprocating force, and creates new unbalance in other direction

Fig. 42—Below—Two counterweights revolving in opposite directions cancel primary reciprocating force

adding another slider which reciprocates in opposite phase, *Fig.* 39. This may take the form of bobweights, *Fig.* 40, or it may be possible to design the machine so as to incorporate a duplicate mechanism. Usually, however, such an expedient is not practicable and it is necessary to confine counterbalancing efforts to simple revolving weights.

Considering for the moment only the primary unbalance, $F_1 cos_\theta$, which is a periodic force of maximum value F_1 and frequency equal to the shaft speed, acting along the line of action of the slider, it will be evident that a revolving counterweight designed to oppose it would introduce a new unbalance $F_w sin\theta$ in the perpendicular

plane, *Fig.* 41. However, reduction of the force in the x-direction at the expense of an additional force in the y-direction may be, and often is, an acceptable solution. For example, if the counterweight produces a force $F_1/2$, the primary is thereby reduced by half while the added force in the y-plane would be equal to only half the primary. This is called half balance.

Complete elimination of the primary force can be effected if, in addition to the counterweight producing $F_1/2$ on the crank, an equal counterweight revolving in the opposite direction is added, *Fig.* 42. Such an expedient would, of course, require the additional complication of a geared shaft. With the two oppositely revolving counterweights in proper phase the vector sum of their forces always completely cancels the primary, leaving no residual primary force in either the x or y-direction. This will be evident from *Fig.* 42.

Secondary Unbalance

Secondary unbalance, $F_2 cos 2\theta$, is a periodic force with a frequency twice that of the crankshaft, hence cannot be even partially counterbalanced by a weight attached to the crankshaft. However, a pair of geared counterweights revolving in opposite directions at twice the crankshaft speed and each producing a force $F_2/2$ would completely cancel the secondary. Similar treatment of the fourth, sixth, etc., order forces would be possible but usually not practical.

Multiple sliders or pistons afford an opportunity to obtain a certain amount of self-balance by placing them in the proper phase relationship, obviating or at least reducing the need for extra counterweights. The chief applications of this principle are in multicylinder engines, compressors, and pumps, and are fully covered in textbooks on balancing and engines (1).

TABLE II

Primary Harmonic

For Coefficient A_1			For Coefficient B_1		
Value of y corrected for sign	Multiplier	Product	Value of y corrected for sign	Multiplier	Product
$+y_0 = 1333$ $-y_{12} = 667$ $2000 \times$	$1.0000 =$	2000	$+y_6 = -222$ $-y_{18} = 444$ $222 \times$	$1.0000 =$	222
$+y_1 = 1284$ $-y_{11} = 648$ $-y_{13} = 706$ $+y_{23} = 1227$ $3865 \times$	$.9659 =$	3733	$+y_5 = 77$ $+y_7 = -441$ $-y_{17} = 655$ $-y_{19} = 137$ $428 \times$	$.9659 =$	413
$+y_2 = 1088$ $-y_{10} = 644$ $-y_{14} = 754$ $+y_{22} = 978$ $3464 \times$	$.8660 =$	3000	$+y_4 = 429$ $+y_8 = -571$ $-y_{16} = 763$ $-y_{20} = -237$ $384 \times$	$.8660 =$	332
$+y_3 = 786$ $-y_9 = 628$ $-y_{15} = 786$ $+y_{21} = 628$ $2828 \times$	$.7071 =$	2000	$+y_3 = 786$ $+y_9 = -628$ $-y_{15} = 786$ $-y_{21} = -628$ $316 \times$	$.7071 =$	223
$+y_4 = 429$ $-y_8 = 571$ $-y_{16} = 763$ $+y_{20} = 237$ $2000 \times$	$.5000 =$	1000	$+y_2 = 1088$ $+y_{10} = -644$ $-y_{14} = 754$ $-y_{22} = -978$ $220 \times$	$.5000 =$	110
$+y_5 = 77$ $-y_7 = 441$ $-y_{17} = 655$ $+y_{19} = -137$ $1036 \times$	$.2588 =$	268	$+y_1 = 1284$ $+y_{11} = -648$ $-y_{13} = 706$ $-y_{23} = -1227$ $115 \times$	$.2588 =$	30
	Sum =	12001		Sum =	1330
$A_1 = \frac{12001}{12} = 1000$			$B_1 = \frac{1330}{12}$		

Resultant $= \sqrt{A_1^2+B_1^2} \doteq 1006$

Phase angle $= tan^{-1}(B_1/A_1) = tan^{-1}(111/1000) = 6° - 20'$

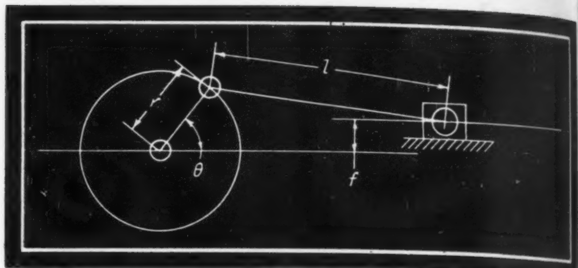


Fig. 43—Above—Offset slider-crank mechanism is example of a nonsymmetrical arrangement, involving phase

TABLE III

Second Harmonic

For Coefficient A_2			For Coefficient B_2		
Value of y corrected for sign	Multiplier	Product	Value of y corrected for sign	Multiplier	Product
$+y_0 =$ $-y_6 =$ $+y_{12} =$ $-y_{18} =$ $\times$	$1.0000 =$		$+y_3 =$ $-y_9 =$ $+y_{15} =$ $-y_{21} =$ $\times$	$1.0000 =$	
$+y_1 =$ $-y_5 =$ $-y_7 =$ $+y_{11} =$ $+y_{13} =$ $-y_{17} =$ $-y_{19} =$ $+y_{23} =$ $\times$	$.8660 =$		$+y_2 =$ $+y_4 =$ $-y_8 =$ $-y_{10} =$ $+y_{14} =$ $+y_{16} =$ $-y_{20} =$ $-y_{22} =$ $\times$	$.8660 =$	
$+y_2 =$ $-y_4 =$ $-y_8 =$ $+y_{10} =$ $+y_{14} =$ $-y_{16} =$ $-y_{20} =$ $+y_{22} =$ $\times$	$.5000 =$		$+y_1 =$ $+y_5 =$ $-y_7 =$ $-y_{11} =$ $+y_{13} =$ $+y_{17} =$ $-y_{19} =$ $-y_{23} =$ $\times$	$.5000 =$	
	Sum =			Sum =	
$A_2 = \frac{Sum}{12} =$			$B_2 = \frac{Sum}{12} =$		

Departure of the simple slider-crank mechanism from the symmetrical arrangement hitherto discussed introduces a new factor—that of phase. The "offset" design shown in *Fig.* 43 affords an example of what occurs in any nonsymmetrical arrangement. In this case the reciprocating force is given by an equation of the form

$$F_{rec} = \frac{W_p + cW_r}{g} \omega^2 r$$

$$[A_1 \cos \theta + B_1 \sin \theta + A_2 \cos 2\theta + etc.] \quad \ldots\ldots\ldots (9)$$

where $A_1 = 1$, $B_1 = f/l +$ etc., and $A_2 = r/l +$ etc. Comparing Equations 8 and 9 it will be noted that in the offset mechanism the primary consists of both cosine and sine terms. What is the resultant and where should a counterbalance be located? Considering the total primary term which is $(A_1 cos\theta + B_1 sin\theta)$, $A_1 cos\theta$ may be

described as the x-component of a vector A_1 which revolves with the crank, *Fig.* 44*a*. The term $B_1 sin\theta$ may be described as the x-component of a vector B_1 at right angles to the crank and revolving with it. Examination of *Fig.* 44*a* will demonstrate the correctness of this concept. From the figure, the resultant is seen to be equal to $\sqrt{A_1^2+B^2}$, the angle which its line of action makes with the crank being equal to $tan^{-1}B/A$. The proper location of a counterweight for partial balance would be as shown in *Fig.* 44*b*.

Inertia couple M_r due to the connecting rod, Equations 3 and 6, is an oscillating moment tending to vibrate the machine about an axis parallel to the shaft centerline. An additional couple results from the acceleration of the slider, the reciprocating force, F_{rec}, having a component along the connecting rod, which exerts a torque on the shaft. An equal and opposite moment on the machine frame results from the component of thrust against the slider guide. The value of the moment due to the inertia of the slider is, approximately,

$$M_p = \frac{W_p}{g}\omega^2 r^2 \left(\frac{1}{4n} sin\,\theta - \frac{1}{2} sin\,2\theta - \frac{3}{4n} sin\,3\theta\right) \quad ..(10)$$

The total moment is $Mr + Mp$.

If the driving and driven units of the machine are integral, the foregoing inertia moment is balanced within the machine. However, if the units are separately mounted or the driven unit is a propeller or the like, the inertia torque must be dealt with by suitable vibration mountings. Also, in such cases there is the additional factor of variations in the driving or driven torque, discussed later in the article.

TABLE IV

Third Harmonic

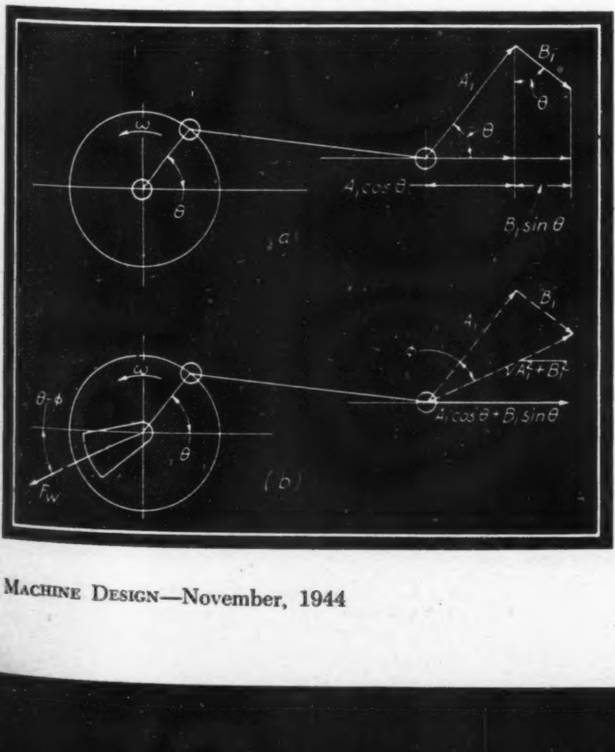
For Coefficient A_3			For Coefficient B_3		
Value of y corrected for sign	Multiplier	Product	Value of y corrected for sign	Multiplier	Product
$+y_0 =$ $-y_4 =$ $+y_8 =$ $-y_{12} =$ $+y_{16} =$ $-y_{20} =$ —— ×	1.0000 =		$+y_2 =$ $-y_6 =$ $+y_{10} =$ $-y_{14} =$ $+y_{18} =$ $-y_{22} =$ —— ×	1.0000 =	
$+y_1 =$ $-y_3 =$ $-y_5 =$ $+y_7 =$ $+y_9 =$ $-y_{11} =$ $-y_{13} =$ $+y_{15} =$ $+y_{17} =$ $-y_{19} =$ $-y_{21} =$ $+y_{23} =$ —— ×	.7071 =		$+y_1 =$ $+y_3 =$ $-y_5 =$ $-y_7 =$ $+y_9 =$ $+y_{11} =$ $-y_{13} =$ $-y_{15} =$ $+y_{17} =$ $+y_{19} =$ $-y_{21} =$ $-y_{23} =$ —— ×	.7071 =	
$A_3 = \frac{Sum}{12} =$	Sum =	══	$B_3 = \frac{Sum}{12} =$	Sum =	══

TABLE V

Fourth Harmonic

For Coefficient A_4			For Coefficient B_4		
Value of y corrected for sign	Multiplier	Product	Value of y corrected for sign	Multiplier	Product
$+y_0 =$ $-y_3 =$ $+y_6 =$ $-y_9 =$ $+y_{12} =$ $-y_{15} =$ $+y_{18} =$ $-y_{21} =$ —— ×	1.0000 =				
$+y_1 =$ $-y_2 =$ $-y_4 =$ $+y_5 =$ $+y_7 =$ $-y_8 =$ $-y_{10} =$ $+y_{11} =$ $+y_{13} =$ $-y_{14} =$ $-y_{16} =$ $+y_{17} =$ $+y_{19} =$ $-y_{20} =$ $-y_{22} =$ $+y_{23} =$ —— ×	.5000 =		$+y_1 =$ $+y_2 =$ $-y_4 =$ $-y_5 =$ $+y_7 =$ $+y_8 =$ $-y_{10} =$ $-y_{11} =$ $+y_{13} =$ $+y_{14} =$ $-y_{16} =$ $-y_{17} =$ $+y_{19} =$ $+y_{20} =$ $-y_{22} =$ $-y_{23} =$ —— ×	.8660 =	
$A_4 = \frac{Sum}{12} =$	Sum =	══	$B_4 = \frac{Sum}{12}$	Sum =	══

Fig. 44—Below—Primary reciprocating force in offset mechanism is sum of components of two revolving vectors



Vibration in Constant-Speed Machines

On machines which operate at an absolutely fixed speed, such as those driven by synchronous motors, it may be possible to employ a dynamic vibration absorber to diminish the effect of unbalanced forces or moments. Consisting essentially of a spring-mounted weight attached to the machine frame and having freedom of movement in the direction of the vibrating force, the absorber is tuned to the frequency of the major disturbance, which might be the primary, secondary or other harmonic. The absorber weight, w, and spring stiffness, k pounds per inch, therefore should be so proportioned that $f = 3.13\sqrt{k/w}$, where f is the frequency of the disturbing force, cycles per second (Equation 1, Part II). At this frequency the weight vibrates on the spring in such a direction that it exerts on the machine frame—through the spring—a force which opposes the disturbance. To be effective the absorber mass evidently must

have freedom to vibrate with an amplitude at least equal to F_n/k, where F_n is the amplitude or maximum value of the disturbing force (F_1, F_2, etc.). Substituting the value of k from the previous expression it is found that the absorber amplitude will be $9.8F_n/wf^2$. This indicates that if the absorber amplitude is to be kept within reasonable bounds, the absorber weight w may have to be substantial, unless the frequency f is high. It should be pointed out that the addition of such an absorber to a resiliently mounted machine creates a system with "two degrees of freedom" and two critical speeds. Therefore, before incorporating an absorber in a machine the designer should explore fully the dynamic characteristics of the setup (see Ref. 2).

Calculations for Complex Mechanisms

For complex mechanisms, including compound linkages as well as cams of noncircular outline, the mathematical expressions for acceleration become extremely involved. In such cases the best approach is to draw the mechanism to scale in a series of positions and to determine the required accelerations of the moving parts by graphical methods such as are described in References (3) and (4). The series of values for acceleration may then be plotted in the form of a curve such as *Fig.* 45.

For purposes of counterbalancing and the design of vibration mountings, however, it is necessary to know the frequencies and magnitudes of the harmonic components of the curve (see Part I, *Fig.* 2). By means of "harmonic analysis" these components can be found. Machines are

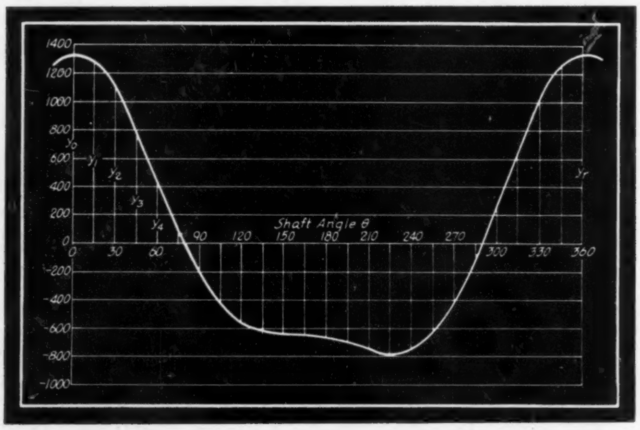


Fig. 45—Acceleration, force, or torque in complex mechanism is plotted for complete revolution of main shaft. Breakdown into component harmonics is effected by scaling ordinates and performing routine calculation with aid of tables

available for performing the necessary calculations, but for the information desired by the engineer the following method of manual computation is entirely satisfactory.

It being understood that any periodic force or other function can be regarded as composed of a series of simple harmonics, the equation to the curve in *Fig.* 45 may be written

$$y = A_1 \cos\theta + A_2 \cos 2\theta + A_3 \cos 3\theta + \ldots + A_n \cos n\theta + \ldots\ldots$$

$$+ B_1 \sin\theta + B_2 \sin 2\theta + B_3 \sin 3\theta + \ldots + B_n \sin n\theta + \quad (11)$$

To find the numerical values of the coefficients A_1, B_1, A_2, B_2, etc., a series of equidistant ordinates is drawn to the curve, as in *Fig.* 45, for a complete cycle or revolution of the main shaft. If there are r divisions the first ordinate ($\theta = 0$ degrees) is called y_o and the last ordinate ($\theta = 360$ degrees) is called y_r. In *Fig.* 45 the number of divisions is 24, the interval between adjacent ordinates being 15 degrees. Each ordinate then is measured and tab-

Table VI

Fifth Harmonic

For Coefficient A_5			For Coefficient B_5		
Value of y corrected for sign	Multiplier	Product	Value of y corrected for sign	Multiplier	Product
$+y_0 =$ $-y_{12} =$ ___ ×	1.0000 =		$+y_6 =$ $-y_{18} =$ ___ ×	1.0000 =	
$+y_5 =$ $-y_7 =$ $-y_{17} =$ $+y_{19} =$ ___ ×	.9659 =		$+y_1 =$ $+y_{11} =$ $-y_{13} =$ $-y_{23} =$ ___ ×	.9659 =	
$-y_2 =$ $+y_{10} =$ $+y_{14} =$ $-y_{22} =$ ___ ×	.8660 =		$-y_4 =$ $-y_8 =$ $+y_{16} =$ $+y_{20} =$ ___ ×	.8660 =	
$-y_3 =$ $+y_9 =$ $+y_{15} =$ $-y_{21} =$ ___ ×	.7071 =		$-y_3 =$ $-y_9 =$ $+y_{15} =$ $+y_{21} =$ ___ ×	.7071 =	
$+y_4 =$ $-y_8 =$ $-y_{16} =$ $+y_{20} =$ ___ ×	.5000 =		$+y_2 =$ $+y_{10} =$ $-y_{14} =$ $-y_{22} =$ ___ ×	.5000 =	
$+y_1 =$ $-y_{11} =$ $-y_{13} =$ $+y_{23} =$ ___ ×	.2588 =		$+y_5 =$ $+y_7 =$ $-y_{17} =$ $-y_{19} =$ ___ ×	.2588 =	
	Sum =			Sum =	
$A_5 = \frac{\text{Sum}}{12} =$			$B_5 = \frac{\text{Sum}}{12} =$		

ulated as in Table I, omitting the last ordinate which, of course, is the same as the first. The value of the coefficient A_1 is given by the formula

$$A_1 = \frac{\text{Sum of all values of the product } y\cos\theta}{\text{One-half the number of divisions}} \quad \ldots\ldots (12)$$

In Table I there is therefore incorporated a column of values of $\cos\theta$, then a column giving the product $y\cos\theta$. The sum of this column divided by half the number of divisions is $11998/12 \approx 1000$, which is the value of A_1.

In like manner the value of coefficient B_1 is given by

$$B_1 = \frac{\text{Sum of all values of the product } y\sin\theta}{\text{One-half the number of divisions}} \quad \ldots\ldots (13)$$

Inclusion of a column of values of $\sin\theta$ and then a column for $y\sin\theta$ (Table I) gives the values of B_1, which is seen

(*Concluded on Page* 162)

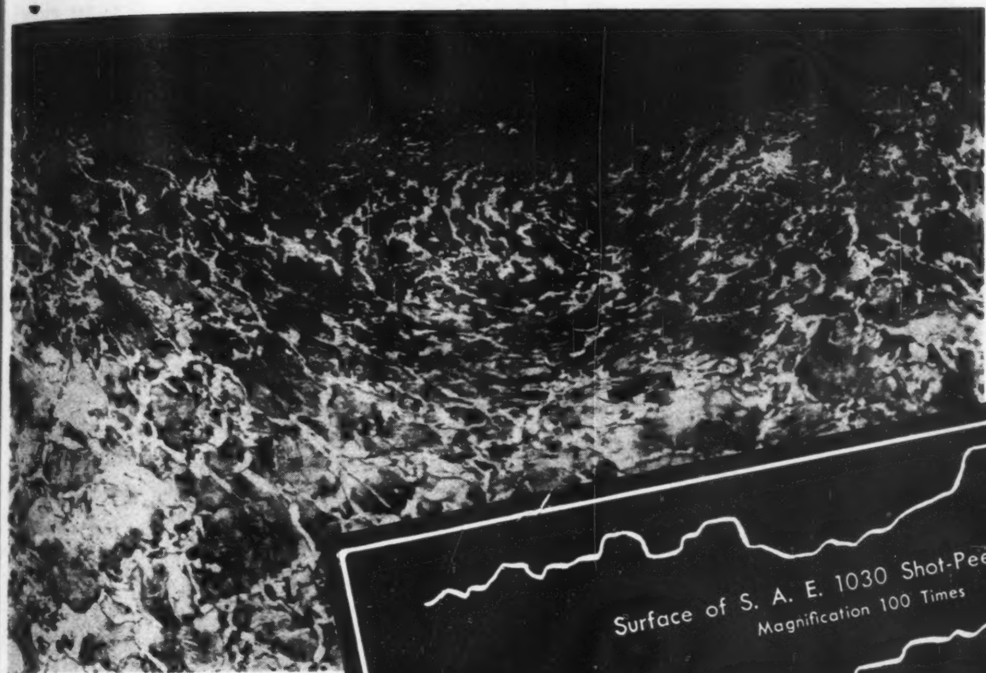


Fig. 1 — Left — Micrograph of SAE 1030 steel shot peened magnification 200. Below — Surface outlines of shot-peened and unpeened steel, traced from micrographs

Effect of Shot Peening on Fatigue Strength

By H. F. Moore
University of Illinois

SHOT PEENING is a term used to denote the process of subjecting the surface of a metal machine part or structural member to a rain of metallic shot driven against the surface by the momentum of the shot as it is released from the rotating blades of a wheel, or by an air blast. As the shot strikes the surface of the part, it produces a shallow layer of metal whose structure, which is made up of crystalline grains, is distorted as shown in *Fig.* 1. This shallow layer of metal is made harder, stronger and less ductile than it was

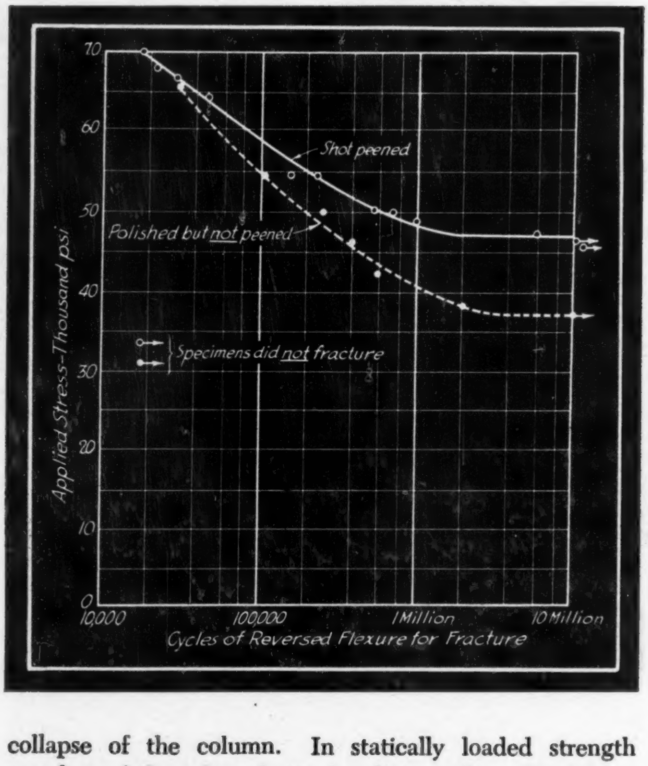
This article is based on a study conducted by the author for the American Foundry Equipment Co.

Fig. 2 — Surface of shot-peened steel, magnified 7 times. No. 19 shot was used, and intensity of peening was to an arc height (h in Fig. 5) .01-inch on an Almen C test strip (See footnote, Page 150)

before it was subjected to the shot-peening process.

Before taking up the discussion of how shot peening strengthens (or, if unskillfully done, may weaken) a metal under repeated stress certain types of structural damage which may be done to structural and machine parts will be noted and briefly described.

Plastic Deformation of a machine part which remains as "set" after load is removed is in itself structural damage in such parts as bolt and nut connections, which become loose when they are loaded beyond the elastic range of the metal. In short and in medium-length columns plastic deformation reduces the stiffness of the metal very greatly and buckling follows, leading to the



collapse of the column. In statically loaded strength members failure by plastic bending under compressive stress (exceeding the yield strength in compression) is probably the most common type of structural damage due to stress.

Shot peening increases the yield strength (resistance to plastic deformation) to a marked degree in the skin of the peened metal, and thus adds strength to resist plastic deformation in the outer layers of a structural part, just where the applied bending stresses are greatest and help is most needed. However, this increase of yield strength may be partly or wholly lost if the peened metal is subjected to subsequent stresses up to or beyond the yield strength of the peened metal. The effectiveness of shot peening in increasing the strength of medium-length columns (an airplane strut, for example) is worthy of further investigation.

Creep is the continuing distortion of a material under a steady load. Damage by creep is similar in character to damage by plastic deformation, but creep continues, while simple plastic distortion does not increase after a few moments of load, or at any rate such increase is negligible. For the common structural metals creep is important only at elevated temperatures. Little is known at present about the effect of shot peening on creep.

Fracture takes place under tensile or shearing stress. While plastic deformation and creep are phenomena involving appreciable volumes of metal before noticeable structural damage is done, fracture may start at some point of localized stress concentration, such as a sharp shoulder on a shaft, and from that nucleus a crack may spread to fracture of the piece. In the case of a brittle metal, fracture may start at such a point under steady load; for ductile metals appreciable yielding takes place under a steady load, and this yielding "evens up" the stress distribution, cutting down the high peaks of stress, with localized plastic deformation rather than fracture. Frequently such localized plastic deformation produces no appreciable structural damage to the part as a whole.

However, under many cycles of repeated stress, especially cycles of reversed stress, a crack may start in the minute area of plastically deformed metal. This crack is due to the exhaustion of the ductility of the overstressed metal, and the crack, once started, will spread, even under a relatively low stress, causing final fracture without appreciable elongation or shortening of the piece fractured. This process of fracture of metals under repeated stress has been given the rather inappropriate name of "fatigue of metals."

Fig. 3—Left—S-N diagrams for SAE 1050 steel, peened and unpeened. Unpeened specimens were given a shop polish

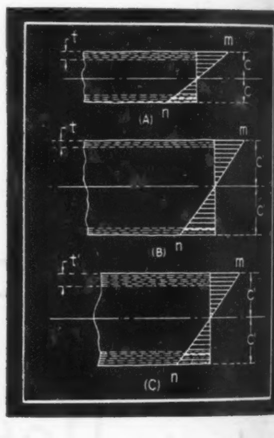


Fig. 4—Right—For a given depth of peening, effectiveness of the peened "skin" will decrease as the stress gradient decreases

Such a spreading crack ("fatigue crack") usually starts under shearing stress within one or more of the crystalline grains which make up a piece of metal. However, as the crack spreads in a region of tensile stress, it soon takes up a direction at right angles to the principal tensile stress. In a region of compressive stress the crack frequently follows a direction of maximum shearing stress, and the crack progresses much more slowly than it does in a region of tensile stress. In structural metals increase of tensile strength is accompanied by increase of shearing strength, and so any process which increases tensile strength usually increases resistance to "fatigue," although not necessarily proportionately.

Shot peening increases the tensile strength of a metal just below the surface, and in addition, sets up a longitudinal compressive stress in the thin "skin" of shot-peened metal. It probably also sets up stresses in a transverse direction and in a direction perpendicular to the surface. Now a fatigue crack nearly always starts on the tension side of a beam, hence the longitudinal residual compressive stress must be overcome by applied stress

before the stress on the tension side of the beam will be a net tensile stress. In effect the beam is strengthened materially against fatigue failure by this layer of metal under residual compressive stress *so long as* that stress is not removed by subsequent working stresses. Hence shot peening acts in two ways to increase the fatigue strength of the metal: (a) The tensile strength of the skin is increased and (b) the effective tensile stress set up by a given applied load is diminished by the compressive stress set up in the skin by shot peening.

Shot peening "cold works" the metal at or near the surface of a piece. Cold work increases the strength of a metal, but uses up part of its ductility, and if *too intense* cold work is done on a metal the ductility may be completely exhausted and a crack started. Then the metal is not strengthened, but is weakened, especially under repeated stress.

How are the microscopic residual stresses set up in the skin of shot-peened metal affected by subsequent applied stresses, especially by thousands of cycles of repeated stress? Test data on this point are few, but tests at the University of Illinois (2)* showed that even a single cycle of applied stress above the yield strength of the peened metal removed the greater part of the residual compressive stress. Repeated cycles of applied stress below the yield strength acted much more slowly, and no serious reduction of the residual compressive stress was found for applied stresses lower than about one third of the yield strength, even though a million cycles of stress were applied.

The question may be asked, "Why should there be *any* reduction of residual stress by cycles of stress lower than the yield strength, or at least by cycles of stress below the 'true' elastic limit?" The answer to this question seems to be that the yield strength is reached at such a stress that plastic deformation has become large enough to do structural damage to the material, and that the existence of any "true" elastic limit for any common structural material is very doubtful. Under any average stress, however low, there are probably localized plastic actions in many isolated crystalline grains of the metal, and under repeated stress there occur microscopic yieldings and a tendency to transfer some stress from locations which have yielded to those which have not, and thus there is a tendency toward a more nearly uniform stress distribution. However, the cumulative action of localized plastic yieldings seems to be almost, if not quite, negligible for stresses less than about one-third the yield strength.

*References in parentheses are listed at end of article.

Rough Peened Surface Not Harmful

At first glance it seems that the rough surface of peened steel must set up localized stress concentrations which might well counteract any gain in fatigue strength due to increase of strength of metal, and to the thin skin of metal under residual compressive stress (*Fig.* 1). However, two factors tend to mitigate the damage due to this rough surface. The first of these factors is the lowering of stress concentration at any one notch or pit by the presence of other notches or pits near by. This has been shown by fatigue tests (3) and by photoelastic analysis of stresses at the base of multiple notches (4). Many notches, or pits, close together seem to share the stress concentration among them, while an isolated notch, or pit, has no such relief from the full stress concentration. Now, as is shown in *Fig.* 2, a shot-peened surface is covered with close spaced pits, and the mitigating factor of multiple pits is present.

Secondly, the maximum stress concentration is at the *bottom* of a pit or a notch, and the action of a rain of hard, spherical shot insures a smooth bottom to the pits, which

TABLE I
Effect of Shot Peening on Fatigue Strength

Metal	Stress Cycle	Endurance Limit for 10,000,000 Cycles of Stress p.s.i. NOT Peened	Endurance Limit for 10,000,000 Cycles of Stress p.s.i. Peened	Unpeened Specimens, Polished	Gain in Fatigue Strength for Peened Specimen, per cent	Reported by
.85% C. Steel Spring Wire	Torsion, 20,000 p.s.i. to maximum	95,000	135,000	NO	42	Zimmerli(5)
SAE 1095 Steel Music Wire	" "	90,000	135,000	NO	50	"
18-8 Stainless Steel Wire	" "	65,000	110,000	NO	69	"
13-2 Stainless Steel Wire	" "	80,000	120,000	NO	50	"
Phosphor Bronze, SAE 81 Wire	" "	35,000	50,000 (c)	NO	43	"
18-8 Stainless Steel	Torsion, zero to max.	46,000	92,000	NO	100	Wahl(6)
NE 9470 Steel, carburized	Reversed Flexure	100,000	152,000	NO	52	H. F. Moore(7)
SAE 4032 Steel, carburized	" "	100,000	150,000	NO	50	"
NE 9420 Steel, carburized	" "	100,000	153,000	NO	53	"
SAE 1020 Steel Plate	" "	34,000	38,000	YES	12	"
SAE 1050 Steel Plate	" "	37,000	47,000	YES	27	"
SAE 1045 Steel, annealed	Reversed Flexure	39,500	43,800	YES	11	Lessells and Murray(8)
SAE 1045 Steel, water quench and draw	" "	80,700	75,000	YES	—7 (a)	"
SAE 9260 Steel, oil quench and draw	" "	108,600	106,000	YES	—3 (b)	"
Armco Iron	" "	26,800	27,000	YES	1	"
SAE X4340 Steel, oil quench and draw	" "	66,000	78,000	YES	18	"
SAE 1045 Steel, normalized and tempered	Reversed Flexure	31,000	32,000 (d)	YES	3	Horger and Neifert(9)
	" "	31,000	34,000 (e)	YES	10	"
	" "	31,000	37,000 (f)	YES	19	"

(a) Residual stresses due to heat treatment may have offset benefits of shot peening. See author's closure in reference.
(b) Peened specimens showed much longer life under stresses above the endurance limit.
(c) For phosphor bronze there may be a reduction of endurance limit under more than 10,000,000 cycles. However, there is shown a distinct increase of strength due to peening.
(d) Peened with No. 28 shot. (e) Peened with No. 19 shot. (f) Peened with No. 22 shot.

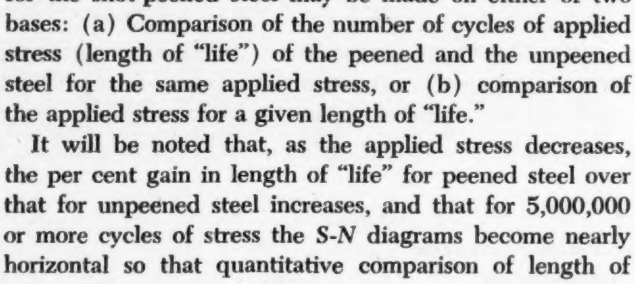
is a further mitigation of stress concentration. Data are not available to enable any quantitative estimate of the effective stress concentration on a shot-peened metal surface. However, the available fatigue test data show that in most cases the net effect of shot peening properly carried out on steel is to increase the fatigue strength in bending above that of polished unpeened specimens.

One caution is given as to stress concentration due to notches caused by shot peening. Thin fins of shot-peened steel may be pushed out at the corners of a sharp-cornered rectangular bar. These fins have edges so jagged and notches so sharp at the bottom that, even with multiple notches along the edge, the stress concentration and the tensile stresses set up under the pushed-out fins may be severe enough to offset completely the advantages of increased strength of metal and of the thin skin of surface metal under compressive stress. These sharp fins may be avoided by chamfering the edges of a rectangular bar or plate before peening it.

Fatigue strength of a metal or of a metal part is evaluated from the results of a series of tests of specimens (or, better yet, of full-size parts) subjected to a series of applied stresses of known magnitude. The variables in a series of fatigue tests are magnitude of applied stress (S) and number of cycles of stress (N) to cause fracture. *Fig.* 3 shows typical S-N diagrams for shot-peened steel and for the same steel not shot-peened. The steel was a rather soft steel, SAE 1050, and the fact that there has been an improvement of fatigue strength in the shot-peened steel is evident at a glance. A quantitative estimate of the percentage of improvement in fatigue strength for the shot-peened steel may be made on either of two bases: (a) Comparison of the number of cycles of applied stress (length of "life") of the peened and the unpeened steel for the same applied stress, or (b) comparison of the applied stress for a given length of "life."

It will be noted that, as the applied stress decreases, the per cent gain in length of "life" for peened steel over that for unpeened steel increases, and that for 5,000,000 or more cycles of stress the S-N diagrams become nearly horizontal so that quantitative comparison of length of

Fig. 5—Test gage measures arc height "h" of a peened test strip, which is an indication of peening intensity



"life" becomes impossible. As the length of "life" increases the gain in per cent of applied stress for the shot-peened metal over the applied stress for the unpeened metal increases and, for this particular steel, becomes 27 per cent when the S-N diagrams are almost horizontal.

The applied stress at which the S-N diagram for a metal becomes horizontal is called the endurance limit for indefinitely long "life," or simply the endurance limit. On account of the impossibility of comparing length of "life" of two metals for applied stresses below the endurance limit of either one of them, the comparison of stresses for a given "life" is to be preferred to the comparison of "life" for a given applied stress. There are, however, many cases in which length of endurance for satisfactory service is a more convenient basis for evaluating resistance to fatigue than is applied stress. An examination of the S-N diagrams shown in *Fig.* 3 shows that a small percentage increase in applied stress (or load) means a much larger percentage decrease in length of service before fracture. Examination of many hundreds of S-N diagrams confirms this statement, and also shows that there is a considerable amount of "scatter" of results even in carefully conducted fatigue tests. This means that whether applied stress or length of "life" before probable fracture is used as the basis of computation by the designer of machine parts, a factor of safety (better called a factor of uncertainty) must be used.

Designing for "Life" or "Load"

If it is desired to design a machine part on the basis of a satisfactory length of service before fracture the designer should use for his computation a "life" many times (say from 10 to 20 times) the desired length of satisfactory service. On the other hand, if the applied stress (or load) for the desired length of satisfactory service is used, the reduction factor for that applied stress, based on the endurance limit for a satisfactory "life," after allowance has been made for stress raisers, residual stresses, size of part, range of applied stress during a cycle, size of piece and any other factors which may be judged to affect fatigue strength—this reduction factor then may be much smaller (say from 1.5 to 3).

TABLE I summarizes results of comparative fatigue strength of peened and unpeened specimens of various metals. The table is based on the applied stress for a "life" of 10,000,000 cycles of stress. For most steels this stress may be regarded as the endurance limit for indefinitely long life; for the phosphor bronze specimens the endurance limit listed in TABLE I is probably somewhat higher than the endurance limit for indefinitely long life.

From TABLE I a comparison can be made of the tests in which the unpeened specimens were polished with the tests in which the unpeened specimens were *not* polished. The percentage gain in strength due to peening is naturally greater for those tests in which the unpeened specimens were not polished, but left with their "as received" surface finish. Examining the tests in which the unpeened specimens were polished it is seen that in eight of the ten items (items marked "YES" in the fifth column of TABLE I) the fatigue strength of the peened specimens was greater than that of the unpeened polished specimens, the percentage of gain in fatigue strength ranging

from 1 to 27 per cent. In two cases the fatigue strength of the polished, unpeened specimens was higher than that of the peened specimens, the percentages being 3 and 7 per cent respectively.

An examination of the detailed reports of these two tests brings out the fact that in the case of the SAE 1045 steel, water quenched and drawn, it was recognized that the heat treatment might have set up residual stresses which acted to counteract the strengthening effect of the peening. In the case of the SAE 9260 steel, oil quenched and drawn, while the fatigue strength at 10,000,000 cycles is slightly less for the peened specimens than for the polished, for applied stresses above the endurance limit for 10,000,000 cycles, the strength of the peened specimens for a given number of cycles of stress was greater than that of the polished specimens. In other words the *S-N* diagram for the peened specimens crossed the *S-N* diagram for the polished specimens at about 10,000,000 cycles of stress.

Offsetting Stress Concentrations

Shot peening has been successfully used at fillets and grooves to offset, by the added strength given to the metal, the stress concentration at the fillet or groove. Shot peening is especially effective in cases in which the shaft, or other part, is not polished, or cannot keep a polished surface. In shot peening grooves and notches (such as screw threads) it is obviously necessary to use shot smaller in diameter than the diameter of the rounded bottom of the groove or notch.

If a machine part is to be subjected to repeated cycles of direct axial tension and/or compression, shot peening is less effective in increasing its fatigue strength than is the case for a part subjected to repeated cycles of bending or of torsion. In a piece subjected to axial tension or compression, up to the yield strength of the core, the applied stress is uniformly distributed over the cross-section, and the bulk of the load must be carried by the unpeened core. At the yield strength of the core yielding takes place without much additional stress being taken by the core, although some of the stress is transferred from core to peened skin, which has a higher yield strength than the core. However, the area of cross-section of the core is very much greater than that of the cross-section of the peened skin, and the help which the core can get from the peened skin is not very great.

In a machine part under bending or torsion the stress varies from zero at the center of gravity of the cross-section to a maximum at the surface. Then the peened skin of the piece resists a larger proportion of the bending moment than does an equal area of metal in the core, and adds a larger percentage to the strength of the piece than that added by the peened skin of the piece in direct axial tension or compression. Under bending or torsion the peened metal adds strength just where it will do the most good.

In *Fig.* 4 (A) is shown a shot-peened piece with thickness $2c$ and thickness of shot-peened skin t. *Fig.* 4 (B) shows a piece with thickness $2c'$ with $c' = 2c$, and the same skin thickness t as in *Fig.* 4 (A). *Fig.* 4 (C) shows a piece with thickness $2c'$ and skin thickness t' which is equal to $2t$. It seems probable that for pieces to resist

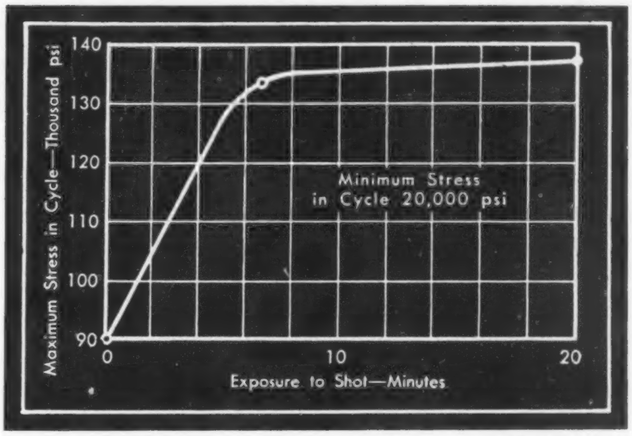


Fig. 6—How time of exposures to the rain of shot affects fatigue strength, for a life of several million cycles

flexure or torsion the increase of fatigue strength due to shot peening will be some function of the ratio of thickness of shot-peened skin to thickness of the piece. It can not be stated at the present time that the piece shown in *Fig.* 4 (C) will resist just as high repeated stresses as will the piece shown in *Fig.* 4 (A), but there will be a tendency to equalize resistance to repeated flexural or torsional stresses if the thickness of the shot-peened skin is increased as the thickness of the piece is increased.

Intensity of shot peening depends on the size of shot, material of shot, striking velocity of shot and length of exposure of the peened surface to the rain of shot. At present no quantitative rules for assigning values to these factors so as to produce optimum results of peening can be given. It seems reasonable to assume that for deep penetration of peening larger shot should be used than for shallow penetration.

Measuring the Intensity of Peening

A useful device for measuring intensity and securing uniformity of peening is the Almen strip gage, shown in *Fig.* 5 (10). A thin flat strip of rather hard steel is clamped to a base and peened for a given time with the same combination of size of shot, material of shot, and striking velocity of shot as is to be used in the peening of a structural or machine part. In fact the test strip is attached to a dummy of the same shape and material as the pieces to be peened. The intensity of peening, therefore, is representative of that given to the surface of the peened part. A time-intensity curve for a batch of pieces may then be plotted from the results of tests made in this manner. After the exposure to the rain of shot for a given time the strip is removed from the base and is found to be curved, with the convex surface on the peened side. This is due to the stresses set up by the peening of one side. The curvature of the peened test strip is taken as a measure of the intensity of the stresses set up by the peening and hence as a measure of the intensity of peening. The distance h in *Fig.* 5, called the "arc height," is then a measure of intensity of peening. The choice of desired intensity of peening for optimum results is, at present, a matter for experience and judgment on the part of the operator of the peening process. The curvature of the Almen strip gage under a given peening tech-

nique depends on (a) degree of coverage of the surface (saturation) and (b) depth of the stressed layer. There is probably some change in hardness as the depth is increased, but this is probably very slight.

In *Fig.* 6, from data reported by Zimmerli (5), is shown the effect of time of exposure on fatigue strength, determined on the basis of applied stress for a "life" of several million cycles of stress. This diagram has a shape similar to that of the intensity-time diagrams previously referred to.

In *Fig.* 7 from Horger and Neifert's test results (9) is shown the effect of peening intensity on fatigue strength. For the plain specimens tested (specimens without fillets) the maximum fatigue strength was obtained with a peening intensity indicated by an arc height of .0065-inch on an Almen "C" strip†. However, for specimens with fillets the fatigue strength increased up to an intensity of .0100 on an Almen "C" strip.

It seems that peening beyond a certain intensity weakens the resistance of a part to repeated stress. Probably this effect of "overpeening" is due to cracks started in the peened surface of the metal, or just below the peened skin. At the present time experience, developed judgment and records of performance in service of parts similar to those to be peened constitute the best basis for determining the optimum intensity of peening for any given batch of parts.

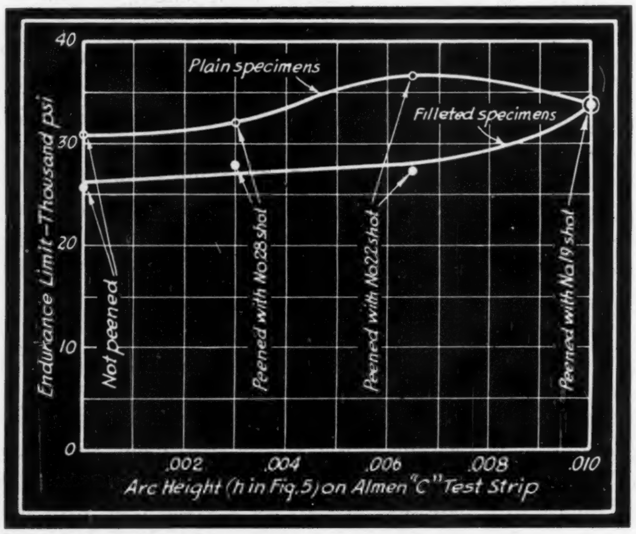


Fig. 7—How the intensity of peening affects fatigue strength of plain and filleted specimens

Shot peening does not directly affect the chemical attack of moisture on steel. However, the surface of shot-peened metal is under residual compressive stress, and before corrosion pits can form and start a fatigue failure, this peened and compressed skin resists cracking and delays, if it does not inhibit, the formation of the corrosion pits, which are the cause or the nuclei for possible spreading cracks. In this matter of corrosion as in the case of optimum intensity of peening, much more research work needs to be done. However, experience has shown that shot-peened steel is distinctly more resistant to destructive corrosion than unpeened steel of the same chemical composition.

Much has yet to be learned about the details of the peening process before any very definite rules can be laid down for securing optimum results for a given machine part. The thickness of the shot-peened "skin" on a machine or structural part is probably not over .01-inch for small pieces, although it may be somewhat larger for large pieces peened with large shot with a high striking velocity. This means that shot peening cannot be expected to be so effective in increasing the fatigue strength of thick pieces as it has been found to be in the case of thin pieces. If there are cracks or seams in the base metal peening may not be effective.

Where Shot Peening Can Be Applied

Shot peening can be applied to irregular shapes, in which heat-treating processes might cause excessive distortion, and in which rolling or drawing processes are not feasible. Shot peening can be applied to finished parts, such as springs, or to specific areas on structural and machine parts, as when shot peening is applied to the fillet of a shaft to offset stress concentration, or to the body of a shaft to resist pitting corrosion. It can be applied to parts such as gear teeth without causing appreciable distortion, while it produces a surface with improved resistance to pitting corrosion under localized heavy stress at bearing points, and also with improved resistance to wear. Often shot peening can be used as a surface finish in place of polishing, with actual gain in fatigue strength. Such a substitution of shot peening for polishing often reduces production costs materially. It is being used with apparent success to increase resistance to surface damage, such as light bruises, "fretting corrosion," pitting corrosion and decarburization. It seems certain that the field of usefulness of shot peening will be much enlarged in the near future.

The author wishes to thank the following organizations and individuals for furnishing data used in this article: Society for Experimental Stress Analysis; American Society for Testing Materials; Messrs. O. J. Horger and H. R. Neifert (Timken Research Laboratories); and Mr. R. E. Peterson (Westinghouse Research Laboratories). Especial acknowledgment is made to Mr. J. O. Almen (General Motors Research Laboratories) who read the first draft and who made many valuable suggestions and criticisms.

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†Two test strips are in use for intensity determinations. For low intensities the "Almen A" strip is used; this strip is ¾ inch wide, 3 inches long and .051-inch thick (±.001-inch). For high intensities the "Almen C" strip is used. This strip is ¾-inch wide, 3 inches long and .0938-inch thick (±.001-inch). Both strips are made of steel with a rockwell "C" hardness of 44-50.

Shop Technique Claims Strict Attention in Present-Day Design

WAR production figures have proved without question that this country has no equal in rapidity of production. Cutbacks that recently have been put into effect, though in some cases called for by the changing conditions of war, attest to the overwhelming speed at which programs have been met. Could these goals have been reached as quickly or as well had it not been that designers as well as production men have taken full advantage of every new method of production that has become available?

The designer with a good all around knowledge of production technique and shop problems has more than proved his worth during these war years. As has been recognized, too many pitfalls await the passage of a new design through the shop for any designer to develop his machine from the standpoint of anticipated performance while neglecting the all-important questions of ease of production and facilities available.

It is not by any means a simple matter for a designer to keep himself intimately posted on all new production methods, particularly during a period of intensive progress and development such as the present. Yet that, in increasing measure, is his job. His horizon is broadening to the point that he must be in a position to appraise the merits of, for example, such processes as shot peening, precision casting, induction heating, high-speed machining, and electrostatic painting—to mention a few of the newer additions.

That these processes are being accorded due recognition, not only by production men but by designers and engineers in general, is evidenced by the record attendance at the recent National Metal Congress and War Conference Display held in Cleveland. Although primarily—as its name implies—an exposition of all that is new in metals, much greater emphasis was placed this year on shop processes and production methods.

The quest for knowledge of production on the part of the designer, and the application of this knowledge in design work, have—as indicated—played an important part in the past. They are destined to come even more closely into the picture as the newer techniques become fully appreciated and more widely adopted.

L. E. Jermy

Outstanding Designs

Laboratory Mill

Frame of this rubber and plastics laboratory mill is of one-piece construction, semisteel. Rolls are hardened steel and their speed adjustment, effected by means of the hand wheel, is infinitely variable throughout a surface-speed range of 36 to 72 feet per minute. The tachometer shown mounted at the top right serves to keep the operator informed at all times of the roll speed.

Manufactured by Erie Engine and Manufacturing Company, this machine is self-contained with motor and starter inside the housing. Cross-rod unit at the top is a safety trip which when actuated disconnects the motor from the power line and sets a magnetic brake. Adjusting screws fitted with calibrated dials are provided for manual alignment of the rolls.

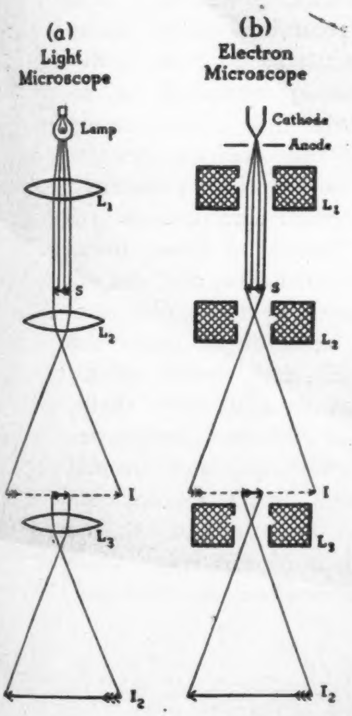
By removing four cap screws the entire housing can be lifted off the base. For routine servicing, the interior of the mill is made accessible by lifting off the end covers. Journal boxes are of semisteel with special bronze bushings.

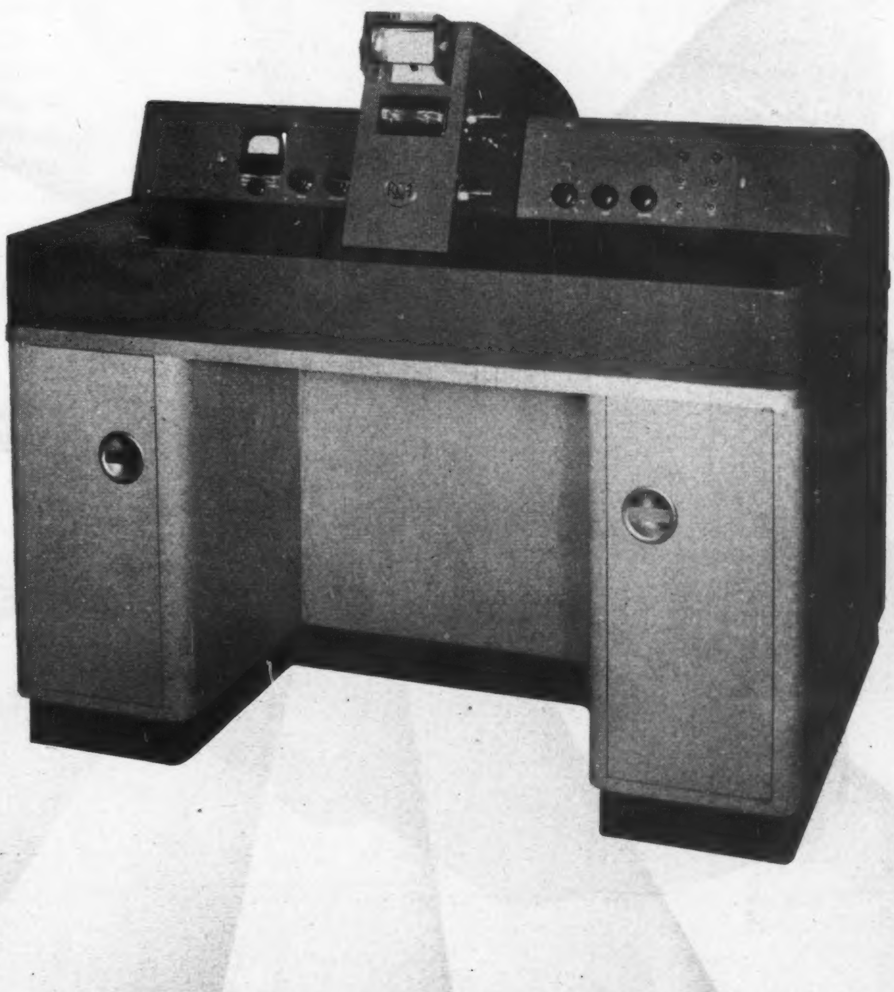
Electron Microscope

Completely self-contained and built to resemble a desk, this electron microscope manufactured by Radio Corp. of America consists essentially of a metal cabinet which encloses the microscope proper, high-voltage and control circuits, and a pumping system. The microscope proper is a large vacuum tube about sixteen inches long, continuously pumped to maintain the vacuum necessary for operation.

Referring to the diagram, in an optical microscope (*a*), light rays from the lamp are formed into a parallel beam and directed on the specimen S by the condenser lens L_1. Image of the specimen then falls on the objective lens L_2 which focuses and magnifies it, producing an enlarged image I. Part of this enlarged image is further magnified by the projector lens L_3. The twice-enlarged image I_2 is that seen by the eye.

In the electron microscope (*b*), source of illumination is a hot cathode which emits electrons. An anode with a small hole in the center gives these electrons a high velocity. A doughnut-shaped coil L_1 produces a field which bends the paths of these electrons, forming them into a parallel beam directed on the specimen S. The electron rays pass through the specimen and are affected in varying degree depending on the composition of the specimen. Those which pass through are brought to a focus by the field of the coil L_2 and form an enlarged image I. The electron rays which form a section of this image are in turn magnified by the field of the coil L_3 and caused to form a further enlarged image I_2. Coils L_1, L_2 and L_3 act like the lenses in an optical microscope, thus are referred to as magnetic lenses.



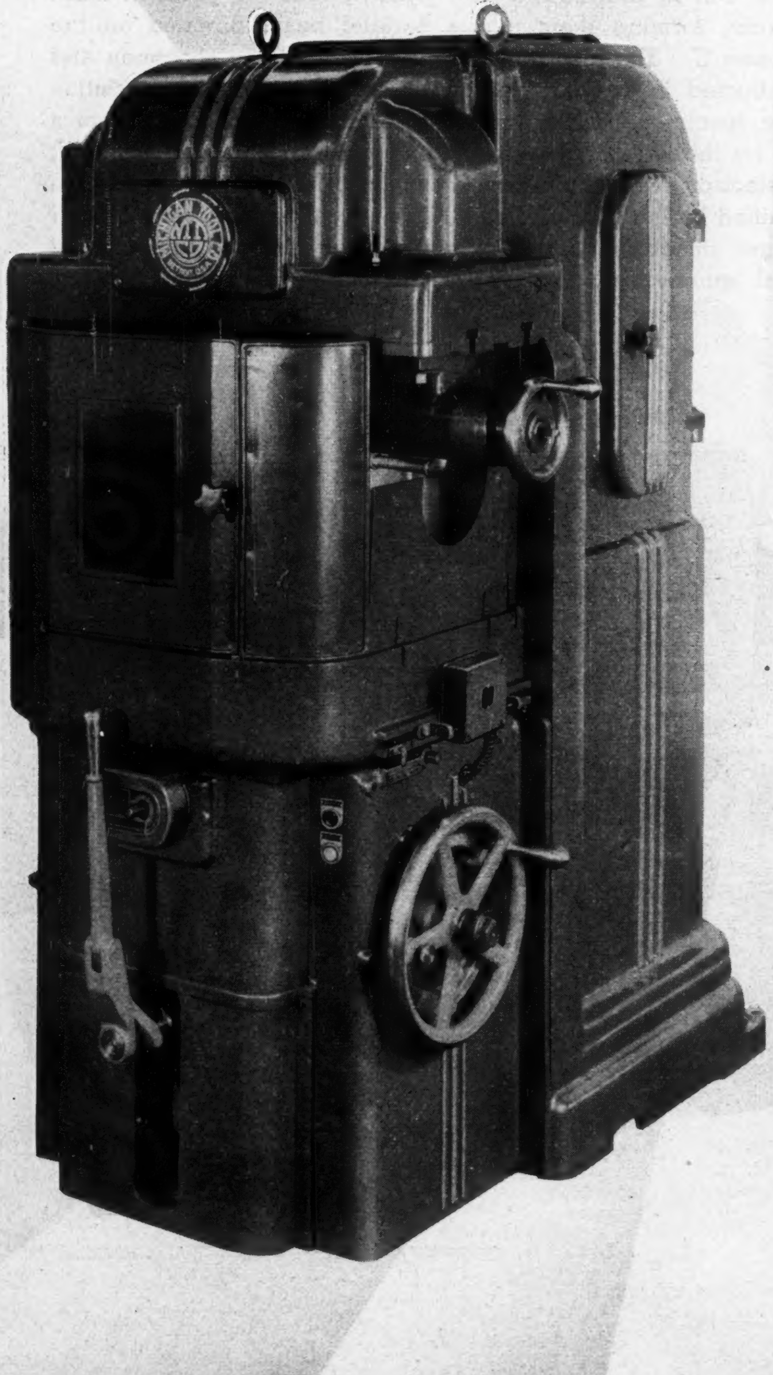


Crossed-Axis Gear Finisher

This machine features "underpass" and "transverse" gear finishing by the crossed-axis shaving method in which the cutter axis is offset from the axis of the work by the amount of helix-angle difference between cutter and work.

Designed and manufactured by Michigan Tool Company, its operation is completely automatic and electrically controlled. Three timing relays and an automatic counter are employed to control the complete cycle of the machine, including infeed, reversal at end of each stroke, number of finishing strokes, etc. Protective releases, if kicked out by overloads, can be reset by pushbutton. Both feed and cutter motors are controlled by reversing starters, the cutter motor being reversible by impulse from one of the timing relays. The electric circuit is fuse-protected throughout and all electrical components are sealed inside a sheet metal cover against dust, dirt and oil.

Cone-drive type alloy-steel pinions and bronze gears are employed and center shafts, cutter shafts and drive shafts, as well as live steel centers, are all of hardened and ground alloy steel. To insure maximum rigidity, the column of the machine is of cast iron. One-shot lubrication is provided.



Aircraft Instrument Tester

Case and turntable of this testing unit formerly were metal. Adapting the design to permit use of plastics in place of the metal entailed various problems in design and in production technique. Upper and lower halves of the housing had to be made sufficiently identical to permit the molding of each plastic half in a single universal mold wherein holes could be plugged or left open as required.

Upper half of the case is a gearbox and center distances between shaft holes required closer tolerances than molding could accomplish. Therefore, brass inserts were molded in and subsequently drilled and reamed to specified tolerances. Alignment of the two base sections, with several gear shafts mounted between them, presented another problem. By snapping one section over a shoulder on the other to withstand side thrust, and adding small dowels in opposite corners to prevent twist, the bearings are kept in accurate alignment.

Markings on the turntable had to be accurately registered but could not be machined because of excessive cost. They are successfully molded by using a phenolic molding compound wherein shrinkage is just enough to permit pulling the part from the mold after a short cooling period. The markings are cut in relief in the walls of the mold and shrinkage of the part in cooling reduces its periphery enough to allow it to clear the raised lettering in the mold.

Development of this unit is the result of the coordinated efforts of The McCaskey Register Co., American Aircraft Associates, Plastic Molding Corp. and Durez Plastics and Chemicals Inc., working in conjunction with U. S. Army engineers.

(New machines listing on Page 200)

Design Roundup

Styling To the Fore

DRESSED-UP APPEARANCE undoubtedly will receive considerable attention when putting on the finishing touches of new design for postwar markets. Lessons learned from experience should go a long way toward preventing recurrences of errors that hindered use or prevented adequate maintenance under the misconception of improved appearance. For instance sweeping covers on occasion have made the operator's position uncomfortable through failure to provide toe or knee holes. Also many enclosures have been welded or otherwise fastened where a hinged panel could have been worked into the design to advantage. More than one motor drive has been totally enclosed without provision for ventilation or maintenance. Such errors require overmotoring and unnecessary expense for motor care. A machine looks good to a customer only if its usefulness is enhanced by "streamlining". For example if cleaning is facilitated, if dark corners are eliminated, if hard-to-get-at recesses (where tools or parts may drop) are avoided, if the appearance is harmonious with the function of the machine, then styling becomes a part of the machine and a definite sales appeal.

Gas Turbines for Autos

ALTHOUGH much current development work is of necessity cloaked in secrecy, there is concerted activity in the field of gas turbines for power plants in the smaller size ranges such as might be suitable for aircraft, and even automobile, installations. Gas turbines have been built for many years in the larger stationary types, and recent studies, together with important metallurgical developments, have indicated tremendous possibilities in the smaller units. In general, the principle of the gas turbine is the combination of a turbine wheel and blower-compressor on the same shaft, the latter compressing the working medium which is then mixed with fuel and exploded, or heated and expanded, against the turbine blades. Two general types are identified as the open and closed-cycle design; in the former the exhaust from the turbine is ejected to the atmosphere, while in the closed type it is recirculated through the compressor. Look for the early announcement of a well-known aircraft engine specialist and a prominent turbine engineer to become associated with a new project in this field.

Sulphite-Treated Alloys

FOUR YEARS of extensive research and test has culminated in sulphite-treated alloy and special steels which provide outstanding machining characteristics without impairment of grain size control, recovery of alloying elements, or physical properties in the lower sulphur ranges. Aside from making possible higher machining speeds, greater accuracy and improved finish, these steels give appreciably longer tool life. Some users, it is reported, have obtained increase in machining production of over 25 per cent, and extended tool life up to 200 per cent. Already, considerable quantities of the steels, developed by the Wisconsin Steel Co., Chicago, have been utilized in crankshafts, camshafts, axles, gears and transmissions, as well as shells. Many specifications have been sulphite-treated: SAE-4140, SAE-5140, NE-9440, NE-9260, NE-8640, WDSS-3, C-1060, C-1046, C-1030, X-1335, etc.

Much Special Engineering Needed

AS AN INSTANCE of the special engineering and tooling that must be done before the automotive industry can get a start toward reconversion, George Romney, managing director, Automotive Council for War Production, cites the case of bushings. Because some of the soft bearing materials are still among the most critical of war materials, ways to conserve them must be utilized in the immediate postwar period.

Instead of the prewar solid bronze bushing with a wall of thirty-six thousandths of an inch, many automobiles will include the new type of bushing with the same thickness, but consisting of a layer of steel under a layer of bronze, so that the bronze is only nine-thousandths thick. This saves 75 per cent of the bearing metal but causes complicated problems—not only production but design and tooling problems as well.

Engineering DATA SHEET

Standard Dimensions for Straight Shaft Serrations

By Frank M. Mallett
Production Design Engineer
Columbus Plant
Curtiss-Wright Corp.

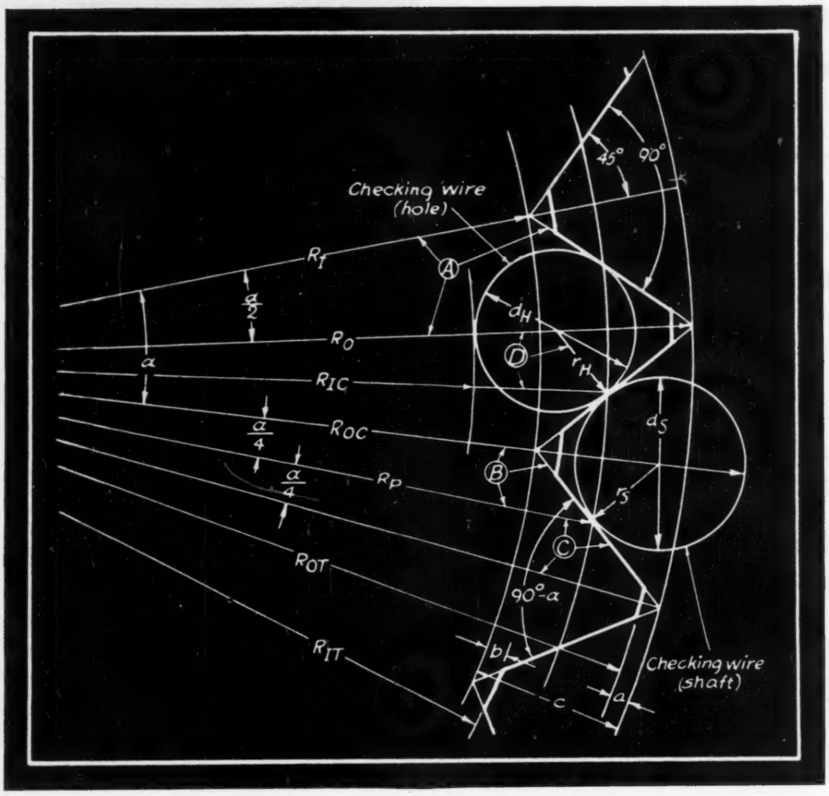
IN THE accompanying Data Sheet complete information on the S.A.E. standard straight shaft serrations is presented in more comprehensive form than hitherto published. This includes formulas for the principal dimensions, TABLE I, design information giving ac-

TABLE I

Formulas for Leading Dimensions

Formula	Dimension	Value of dimension for $N=N$	$N=36$	$N=48$
(1)	α	$\frac{360°}{N}$	10°	7½°
(2)	D_I	$\frac{\sin\left(45° - \frac{\alpha}{4}\right)}{\sin 45°} D_P$	$.9554 D_P$	$.9668 D_P$
(3)	D_O	$\frac{\sin\left(45° - \frac{\alpha}{4}\right)}{\sin\left(45° - \frac{\alpha}{2}\right)} D_P$	$1.0510 D_P$	$1.0368 D_P$
(4)	D_I	$\frac{\sin\left(45° - \frac{\alpha}{2}\right)}{\sin 45°} D_O$	$.9090 D_O$	$.9325 D_O$
(5)	D_{IC}	$\left(\frac{\sin\left(45° - \frac{\alpha}{4}\right)}{\sin\left(45° - \frac{\alpha}{2}\right)} - \frac{\left[1+\csc\left(45° - \frac{\alpha}{2}\right)\right]\sin\frac{\alpha}{4}}{\sin\left(45° + \frac{\alpha}{2}\right)}\right) D_P$	$.9055 D_P$	$.9273 D_P$
(6)	D_{OC}	$\frac{\sin\left(45° - \frac{\alpha}{4}\right) + (1+\sqrt{2})\sin\frac{\alpha}{4}}{\sin 45°}$	$1.1043 D_P$	$1.0784 D_P$
(7)	d_H	$\frac{\sin\frac{\alpha}{4}}{\sin\left(45° + \frac{\alpha}{2}\right)} D_P$	$.0569 D_P$	$.0435 D_P$
(8)	d_S	$\frac{\sin\frac{\alpha}{4}}{\sin 45°} D_P$	$.0617 D_P$	$.0463 D_P$
(9)	a_H	$\frac{1}{2}(D_O - D_{OTH})$		
(10)	a_S	$\frac{1}{2}(D_O - D_{OTS})$		
(11)	b_H	$\frac{1}{2}(D_{ITH} - D_I)$		
(12)	b_S	$\frac{1}{2}(D_{ITS} - D_I)$		
(13)	c	$\frac{1}{2}(D_I - D_O)$		
(14)	e	$\frac{1}{2}(D_{OTH} - D_{ITH}) = \frac{1}{2}(D_{OTS} - D_{ITS})$		
(14′)	e	$c-a-b$, where a and b are either both H or both S		

For list of symbols see Nomenclature on Page 160, also *Fig.* 1.



tual dimensions for standard shafts serrations, TABLE II, reference information for tool designers and engineers, TABLE III, and inspection information, TABLE IV. There is also a brief discussion of the methods of producing serrations, and the effect of these methods on design.

TABLES II, III and IV give all the dimensions needed for standard shaft sizes, while the formulas in TABLE I will enable the designer to extend the range. Formulas 1 to 13 inclusive are derived with the aid of *Figs.* 1 and 2. Inasmuch as most serrations follow the standard, the special formulas for $N=36$ and $N=48$ are more useful. It will be noted that e in Formula 14 is the same for both shaft and hole.

In computing the tables the A.S.A. method of rounding off decimals is used, except in some cases for a and b where they were rounded off in such a way that Formula 14′ agrees with 14.

Fig. 1—Left—Leading dimensions of shaft serrations and checking wires. For triangles A, B, C, and D see Fig. 2

TABLE II
Design Information

Nominal Diameter D_N	No. of Serrations N	Central Angle α	Hole: Pitch Dia. D_P		Hole: Large Dia. D_{OTH}		Hole: Small Dia. D_{ITH}		Shaft: Pitch Dia. D_P		Shaft: Large Dia. D_{OTS}		Shaft: Small Dia. D_{ITS}	
.125	36	10°	.121	+.001 −.000	.125	+.001 −.000	.117	+.001 −.000	.121	+.000 −.001	.124	+.000 −.001	.116	+.000 −.001
.188	36	10°	.181	”	.187	”	.175	”	.181	”	.186	”	.174	”
.250	36	10°	.242	”	.250	”	.234	”	.242	”	.249	”	.233	”
.312	36	10°	.302	”	.312	”	.292	”	.302	”	.311	”	.291	”
.375	36	10°	.362	”	.375	”	.351	”	.362	”	.374	”	.350	”
.500	36	10°	.484	”	.500	”	.468	”	.484	”	.499	”	.467	”
.625	36	10°	.604	”	.625	”	.583	”	.604	”	.624	”	.582	”
.750	48	7½°	.732	”	.750	+.002 −.000	.714	+.002 −.000	.732	”	.749	+.000 −.002	.713	+.000 −.002
.875	48	7½°	.854	”	.875	”	.833	”	.854	”	.874	”	.832	”
1.000	48	7½°	.976	”	1.000	”	.952	”	.976	”	.999	”	.951	”
1.125	48	7½°	1.097	”	1.125	”	1.069	”	1.097	”	1.124	”	1.068	”
1.250	48	7½°	1.219	”	1.250	”	1.188	”	1.219	”	1.249	”	1.187	”
1.375	48	7½°	1.341	”	1.375	”	1.307	”	1.341	”	1.374	”	1.306	”
1.500	48	7½°	1.463	”	1.500	”	1.426	”	1.463	”	1.499	”	1.425	”
1.750	48	7½°	1.707	”	1.750	”	1.664	”	1.707	”	1.749	”	1.663	”
2.000	48	7½°	1.9505	+.0015 −.000	2.000	”	1.902	”	1.9505	+.000 −.0015	1.999	”	1.901	”
2.250	48	7½°	2.1945	”	2.250	”	2.140	”	2.1945	”	2.249	”	2.139	”
2.500	48	7½°	2.4385	”	2.500	”	2.378	”	2.4385	”	2.499	”	2.377	”
2.750	48	7½°	2.6825	”	2.750	”	2.616	”	2.6825	”	2.749	”	2.615	”
3.000	48	7½°	2.9265	”	3.000	”	2.854	”	2.9265	”	2.999	”	2.853	”

For list of symbols see Nomenclature on Page 160, also Fig. 1.

How Production Methods Affect Design

SHAFT serrations can be produced by milling, extruding or external broaching. The latter two are more accurate than the former, and are cheaper for more than a few parts.

Between the serrated portion of the shaft and the plain portion space must be allowed for imperfect serrations due to "runout". In the case of a milling cutter of radius R, the runout will be approximately $\sqrt{2eR}$ where e is the depth of the serration. In the other two methods, about .09-inch is ample. The designer should allow for the larger of these.

If there is a shoulder on the shaft, so that the serrated portion is on a smaller diameter than the adjacent portion, a clearance should be allowed. A minimum of .087-inch is needed for extruding, and this is sufficient also for broach-

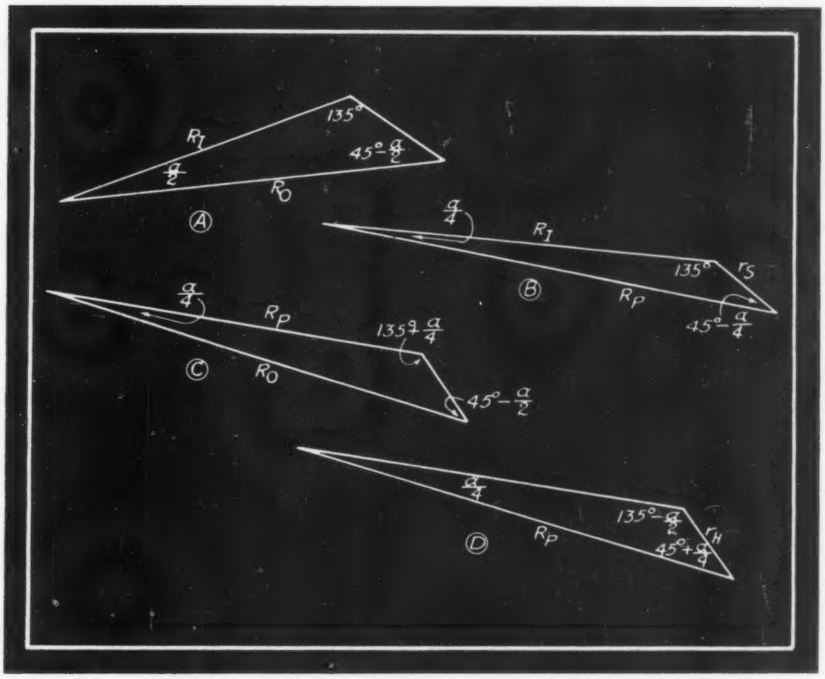


Fig. 2—Triangles from Fig. 1 are used in deriving the formulas which relate the leading dimensions of shaft serrations

TABLE III

Reference Information

D_N	N	α	D_P	D_I	D_O	a_H	a_S	b_H	b_S	c	e
.125	36	10°	.121	.1156	.1272	.0011	.0016	.0007	.0002	.0058	.0040
.188	36	10°	.181	.1729	.1902	.0016	.0021	.0010	.0005	.0086	.0060
.250	36	10°	.242	.2312	.2543	.0022	.0027	.0014	.0009	.0116	.0080
.312	36	10°	.302	.2885	.3174	.0027	.0032	.0017	.0012	.0144	.0100
.375	36	10°	.362	.3459	.3805	.0028	.0033	.0025	.0020	.0173	.0120
.500	36	10°	.484	.4624	.5087	.0044	.0049	.0028	.0023	.0232	.0160
.625	36	10°	.604	.5771	.6348	.0049	.0054	.0029	.0024	.0288	.0210
.750	48	7½°	.732	.7077	.7589	.0045	.0050	.0031	.0026	.0256	.0180
.875	48	7½°	.854	.8256	.8854	.0052	.0057	.0037	.0032	.0299	.0210
1.000	48	7½°	.976	.9436	1.0119	.0060	.0065	.0042	.0037	.0342	.0240
1.125	48	7½°	1.097	1.0606	1.1374	.0062	.0067	.0042	.0037	.0384	.0280
1.250	48	7½°	1.219	1.1785	1.2639	.0070	.0075	.0047	.0042	.0427	.0310
1.375	48	7½°	1.341	1.2965	1.3903	.0077	.0082	.0052	.0047	.0469	.0340
1.500	48	7½°	1.463	1.4144	1.5168	.0084	.0089	.0058	.0053	.0512	.0370
1.750	48	7½°	1.707	1.6503	1.7698	.0099	.0104	.0069	.0064	.0598	.0430
2.000	48	7½°	1.9505	1.8857	2.0223	.0111	.0116	.0082	.0077	.0683	.0490
2.250	48	7½°	2.1945	2.1216	2.2753	.0127	.0132	.0092	.0087	.0769	.0550
2.500	48	7½°	2.4385	2.3575	2.5282	.0141	.0146	.0103	.0098	.0854	.0610
2.750	48	7½°	2.6825	2.5934	2.7812	.0156	.0161	.0113	.0108	.0939	.0670
3.000	48	7½°	2.9265	2.8293	3.0342	.0171	.0176	.0123	.0118	.1024	.0730

For list of symbols see Nomenclature on Page 160, also *Fig.* 1.

ENGINEERING DATA SHEET

ing. If milling is to be used, the clearance c might be given by

$$c = \sqrt{2HR - R^2} + .06$$

where H is the height of the shoulder (difference in radii) and R is the cutter radius.

Design should be such that the clearance area can be undercut if necessary, as in broaching. Diameter of the undercut should be about .01-inch smaller than the inner truncated diameter, D_{IT}. This undercut should be made optional, to allow for more than one way of producing the serration.

It is desirable also to allow a small chamfer at the end of the serrated shaft, about .005-inch or .01-inch more than e, the depth of the serration.

HOLE serrations can be produced by broaching or by cutting with a shaper (for example). The latter obviously is more expensive except for very few parts. If the part with the serrated hole is to fit flush against a shoulder on the shaft, the hole should be counterbored, diameter $D_N + .010$ and depth $c + .010$ where c is the length of the clearance or undercut on the shaft. The value of .010 given here should be thought of as a minimum and could well be increased.

TABLE IV

Inspection Information

D_N	D_P	d_H	d_S	D_{IC}	D_{OC}	Tolerances on 45° Angle
.125	.121	.00688	.00747	.1096	.1336	±2°
.188	.181	.01030	.01117	.1639	.1999	±1½°
.250	.242	.01377	.01493	.2191	.2672	±1¼°
.312	.302	.01718	.01863	.2735	.3335	±1°
.375	.362	.02060	.02234	.3278	.3998	"
.500	.484	.02754	.02986	.4383	.5345	±¾°
.625	.604	.03437	.03727	.5469	.6670	±½°
.750	.732	.03184	.03389	.6788	.7894	"
.875	.854	.03715	.03954	.7919	.9210	"
1.000	.976	.04246	.04519	.9050	1.0525	"
1.125	1.097	.04772	.05079	1.0172	1.1830	"
1.250	1.219	.05303	.05644	1.1304	1.3146	±¼°
1.375	1.341	.05834	.06209	1.2435	1.4461	"
1.500	1.463	.06364	.06774	1.3566	1.5777	"
1.750	1.707	.07425	.07903	1.5829	1.8408	"
2.000	1.9505	.08485	.09031	1.8087	2.1034	"
2.250	2.1945	.09546	.1016	2.0350	2.3665	±10′
2.500	2.4385	.1061	.1129	2.2612	2.6297	"
2.750	2.6825	.1167	.1242	2.4875	2.8928	"
3.000	2.9265	.1273	.1355	2.7137	3.1559	"

For list of symbols see Nomenclature below, also *Fig.* 1.

Nomenclature

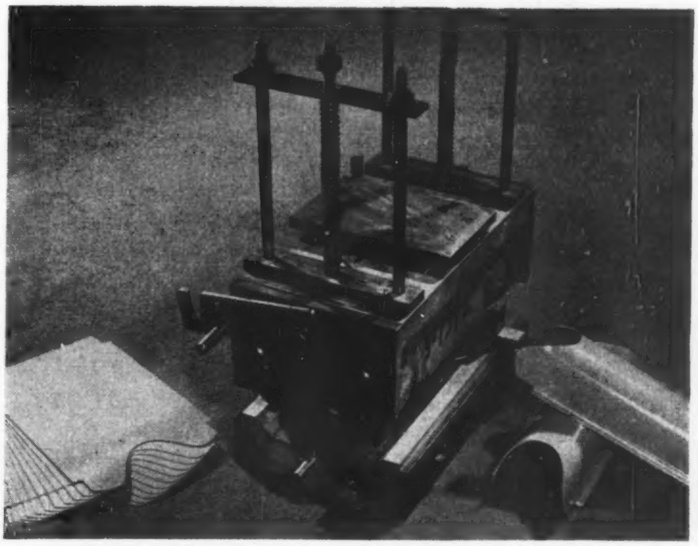
D_N = Nominal diameter = $2R_N$ } Given by SAE standard, see TABLE II
D_P = Pitch diameter = $2R_P$
D_I = Inner diameter = $2R_I$
D_O = Outer diameter = $2R_O$
D_{IT} = Inner truncated diameter = $2R_{IT}$ } Given by SAE standard for both hole and shaft, see TABLE II.
D_{OT} = Outer truncated diameter = $2R_{OT}$
D_{IC} = Diameter across wires (hole) = $2R_{IC}$
D_{OC} = Diameter across wires (shaft) = $2R_{OC}$
N = Number of serrations (Given by SAE standard, TABLE II)
α = Central angle, also angle of adjustment
a_H = Depth of outer truncation, hole
a_S = Depth of outer truncation, shaft
b_H = Depth of inner truncation, hole
b_S = Depth of inner truncation, shaft
c = Depth of serration
e = Depth of truncated serration
d_H = Diameter of checking wire for hole = $2r_H$
d_S = Diameter of checking wire for shaft = $2r_S$

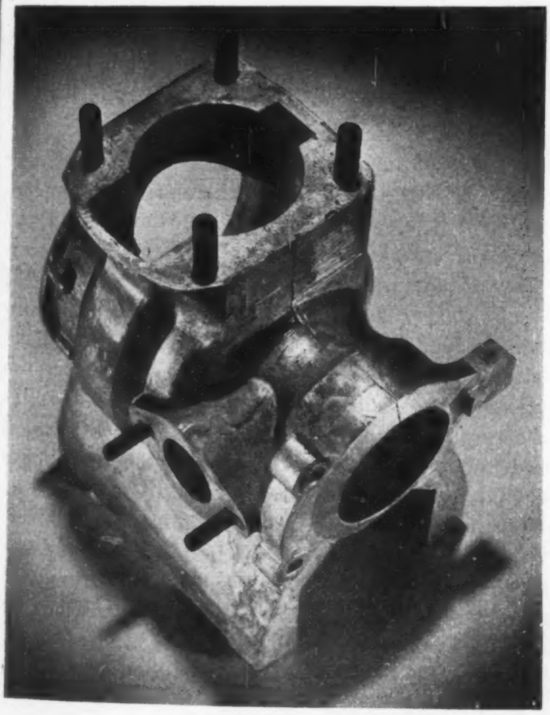
Applications

of Engineering Parts, Materials and Processes

Plastic Laminates Pressed to Shape

TECHNIQUE of producing Formica parts for ammunition chutes on bombers is illustrated at right. Cut blanks of the laminated material, which already has been cured, are reheated to a temperature just below the blistering point and stamped quickly to shape in a forming press employing inexpensive dies made of wood or Pregwood. As an alternative to the earlier method of making such parts in molds, the new development gives promise of useful postwar applications.



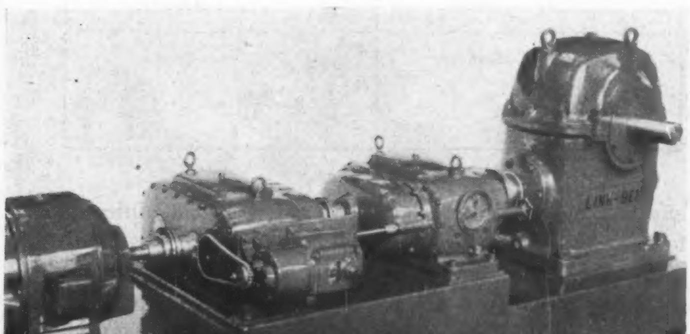


Die-Cast Crankcase for Parachute 'Cycle

ZINC alloy die casting shown at left is the crankcase for the lightweight motorcycle now being used by paratroopers in air invasions. Known as the Servi-Cycle, the machine can be dropped from a plane, utilizing one 24-foot parachute. It also is used in beach landings, in which case it is ridden down the ramp of the landing craft. The crankcase casting is produced in one piece with steel mounting studs molded in as inserts.

Wide Range Variable-Speed Drive

WIDE speed range at extremely low speeds is obtainable through the use of two Link-Belt P.I.V. gear units in tandem, as shown in the conveyor drive at right. The two 6-to-1 ratio units are coupled between an 860-revolutions per minute motor and a worm gear reducer with a 40-to-1 ratio. Used in the chemical industry, the drive provides infinitely variable output speeds ranging from 1/3 to 12 revolutions per minute and incorporates explosion-proof electric remote control.



Vibration and Noise

(*Concluded from Page* 144)

to be $1333/12 \approx 111$.

For higher harmonics a similar procedure is followed. Thus if n is the number of the harmonic, the general formulas for A_n and B_n are

$$A_n = \frac{\text{Sum of all values of the product } y \cos n\theta}{\text{One-half the number of divisions}} \quad \ldots\ldots (14)$$

$$B_n = \frac{\text{Sum of all values of the product } y \sin n\theta}{\text{One-half the number of divisions}} \quad \ldots\ldots (15)$$

If the curve is of simple shape the higher harmonics probably are zero or negligible. To see how closely the terms already calculated fit the actual curve, the values of A_1 and B_1 could be substituted in Equation 11 and a few successive values of y calculated and compared with the actual curve.

The foregoing general method may be adapted to any periodic curve and to any number of ordinates or divisions. The calculations may be systematized and set up in the form of tables such as TABLE I, leaving blank the columns for y and its products, so that unskilled personnel may perform the computations with the aid of a calculating machine.

Improved accuracy and some saving in labor may be effected by arranging the table in somewhat different form. It will be noted that the values of $\cos\theta$, $\sin\theta$, etc., repeat themselves several times. This suggests the form shown in TABLE II, which has been set up for 24 divisions, as in TABLE I. In transcribing values of y from TABLE I to TABLE II it is of course essential that the correct sign be recorded. For example, $y_{12} = -667$, therefore $-y_{12} = +667$. Blank forms for determining all the harmonics up to and including the sixth are presented in TABLES III to VII.

TABLE VII

Sixth Harmonic

For Coefficient A_6			For Coefficient B_6		
Value of y corrected for sign	Multiplier	Product	Value of y corrected for sign	Multiplier	Product
$+y_0 =$			$+y_1 =$		
$-y_2 =$			$-y_3 =$		
$+y_4 =$			$+y_5 =$		
$-y_6 =$			$-y_7 =$		
$+y_8 =$			$+y_9 =$		
$-y_{10} =$			$-y_{11} =$		
$+y_{12} =$			$+y_{13} =$		
$-y_{14} =$			$-y_{15} =$		
$+y_{16} =$			$+y_{17} =$		
$-y_{18} =$			$-y_{19} =$		
$+y_{20} =$			$+y_{21} =$		
$-y_{22} =$			$-y_{23} =$		
$\times$	1.0000 =		$\times$	1.0000 =	
	Sum =			Sum =	
$A_6 = \frac{\text{Sum}}{12} =$			$B_6 = \frac{\text{Sum}}{12} =$		

External driving and driven forces or moments such as gas or liquid pressures, and forces due to the work being performed by the machine, create a periodic torque variation at the main shaft. As in the case of inertia torque previously discussed, this torque variation must be suitably dealt with if the driving and driven elements are not integral. Because it usually is impossible to express the driving or driven forces mathematically, it is necessary to use the method of harmonic analysis to find the magnitude and frequencies of the principal disturbing moments. The general method of procedure is given in the following.

Graphical Method Determines Forces

The mechanism is drawn to scale in a series of positions, for each of which the magnitude, direction and point of application of the external force are known. At any one of these positions the momentary energy input or output is equal to the force F times the velocity v (in the direction in which the force acts) of the point of application of the force. For example, in an engine it would be the product of gas pressure on the piston times the piston velocity. Assuming no losses, the momentary energy at the shaft would be equal to the torque times the angular velocity of the shaft. Equating the two energies, $T\omega = Fv$, from which

$$T = F\frac{v}{\omega} \quad \ldots\ldots (16)$$

where T is the torque or torque reaction to be found. For each position of the mechanism, therefore, the appropriate velocity v is found graphically. Substitution in Equation 16 gives a series of values of the torque, which is then plotted against shaft angle, as in *Fig.* 45. The analysis of the curve follows the procedure already outlined, and is facilitated by the use of TABLES II to VII.

In a few cases, such as in four-stroke cycle engines, two revolutions of the shaft may be required to complete the periodic torque curve. The procedure in such a case is fundamentally the same as before, but the shaft angles will extend from 0 to 720 degrees. With 15-degree intervals, as in TABLE I, there would therefore be 48 divisions. Inasmuch as the lowest frequency is only one-half the speed of the shaft, there will be "half-harmonics" present. This means that in TABLE I there would be columns for $\cos(\theta/2)$, $\sin(\theta/2)$, $\cos(3\theta/2)$, $\sin(3\theta/2)$, etc., as well as the whole-number values such as $\cos\theta$, $\sin\theta$, $\cos 2\theta$, $\sin 2\theta$, etc.

In addition to its importance in connection with the mounting of certain types of machines, the calculation of torque harmonics, as just discussed, is an indispensable step in the analysis of torsional vibration in shafts. The fundamental principles of torsional vibrations and how they may be avoided will be discussed in the next article of this series.

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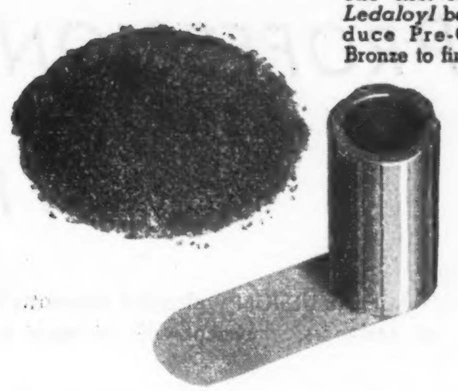
4. A. S. Hall and E. S. Ault—"How Acceleration Analysis Can Be Improved", MACHINE DESIGN, Feb. and March, 1943; also "Auxiliary Points Aid Acceleration Analysis", MACHINE DESIGN, Nov., 1943.

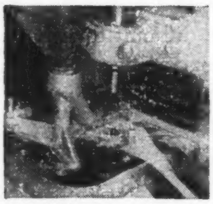
JOHNSON
Ledaloyl
SELF LUBRICATING
BRONZE

The first step in making *Ledaloyl* bearings is to reduce Pre-Cast Bearing Bronze to fine powder.

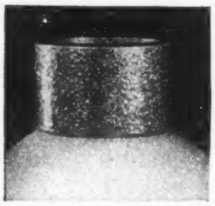


Powder Metallurgy

for the Products of Tomorrow



80,000 pounds per square inch are required to compress LEDALOYL powder into an average size bearing.



Place a LEDALOYL bearing on a lighted lamp bulb. Watch the oil sweat from the pores. Remove the bearing and the oil is reabsorbed. This illustrates the self lubricating and oil retaining action.

Manufacturers planning their postwar products will do well to consider the use of powder metallurgy. It offers many distinct advantages in the way of low cost, light weight, long and dependable service.

Sleeve bearings are a good example. When manufactured from Johnson LEDALOYL, they combine all the good characteristics found in other bearings . . . plus self lubrication. This often enables the designer to eliminate lubrication devices and to seal the bearing in place. When properly designed and installed LEDALOYL outlasts the motive unit in which it is used.

It's an easy matter to determine how LEDALOYL can serve you. Simply call in a Johnson Engineer. Permit him to review your applications . . . to make recommendations based on facts . . . free from prejudice. There is one located as near as your phone. Why not call him—TODAY?

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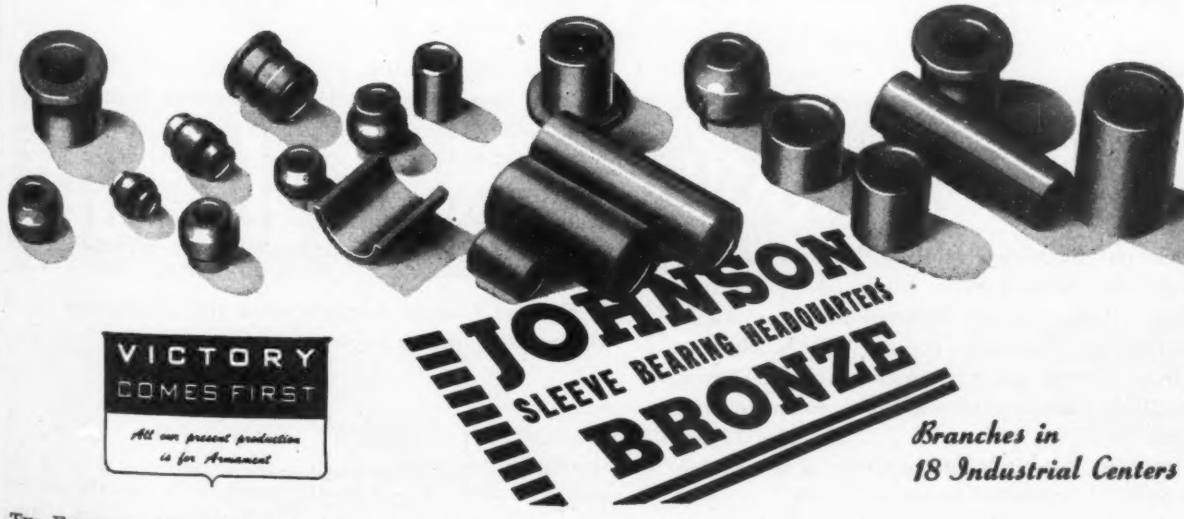

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PROFESSIONAL VIEWPOINTS

MACHINE DESIGN welcomes comments from readers on subjects of interest to designers. Payment will be made for letters and comments published

" . . . die castings in postwar"

To the Editor:

In your July, 1944, issue B. W. Hindman stated on Page 190 that "It is safe to assume that practically all markets for die castings will return. A more extensive use of die castings will undoubtedly be made by design engineers, when there is a demand for a proved engineering material and a method of fabrication capable of accurately reproducing a great number of parts with the minimum expenditure of time, labor and material."

It is believed that the writer of this statement has overlooked two important fields that have been developed during the war. Designers are well aware of the possibilities of using plastics from the artistic standpoint and also to gain the advantages of a nonmetallic material. This field alone will undoubtedly make considerable inroads into the future sales of die castings.

The other development is the rapidly expanding use of powder metals. Many articles are now under construction that will compete with die castings and still further reduce the previous market.

Although it is true that die castings are a great improvement over many other methods of manufacture and will be used considerably in the postwar period, it is inadvisable to state that practically all prewar markets will return. However, it is probable that new applications will be discovered for die castings and it is equally probable that the fields of plastics and powder metals will expand due to postwar activity.

—CHARLES DELMAR TOWNSEND
Process Engineer

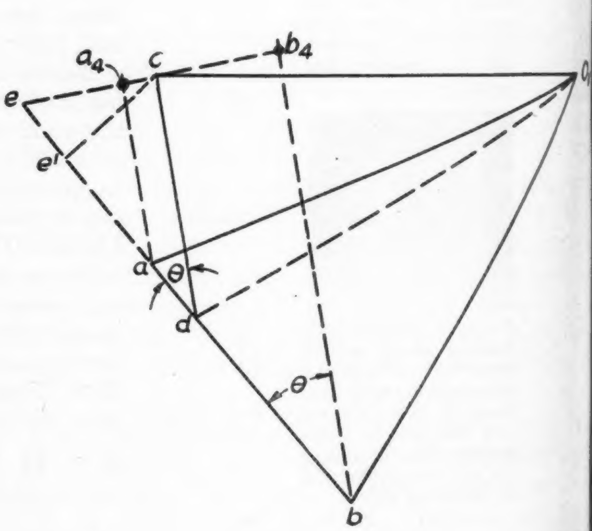
" . . , aid of auxiliary points"

To the Editor:

The article on "Simplified Method Shortens Acceleration Analysis", by Leon Beskin, published in MACHINE DESIGN, June, 1944, uses as illustrations two examples first presented in "Auxiliary Points Aid Acceleration Analysis", by A. S. Hall and E. S. Ault, MACHINE DESIGN, November, 1943. Because of this some comment from us is justified.

In this country the circle of inflection is not widely known and we are fortunate that Mr. Beskin has explained its principle and demonstrated its use so clearly. However, in the solution of the problem shown in his *Fig.* 3 it seems to us that the circle of inflection method is no simplification. It is quite easy to think of a number of circumstances where the method would be more laborious and for many persons more difficult to understand and to execute. Throughout our experience, we have not found any widespread simplification by the circle of inflection method.

Use of one special auxiliary point for the solution of the problem shown in Mr. Beskin's *Fig.* 3 results in considerable simplification.



The imagining of the auxiliary point D as a point on a flexible connector between points A and B eliminates the need for two auxiliary points as used by Hall and Ault. Unfortunately, however, the explanation of the solution is not clear and no mention is made as to the reason for the use of point e. This point e is incorrectly located as the following explanation will make clear.

For the Coriolis component of the acceleration of D the proper expression relative to C is

$$2\omega_4 V_{dc} = 2\frac{V_{a_4b_4}}{AB} V_{dc}$$

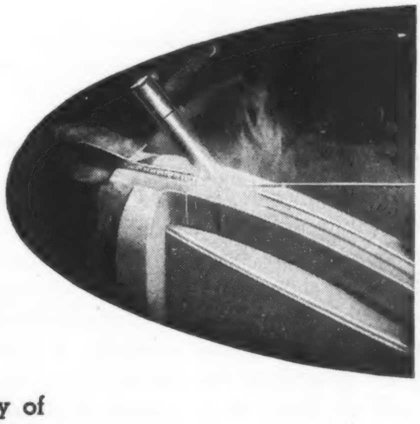
(Continued on Page 166)

WE'VE BEEN ASKED

THIS ABOUT MAGNESIUM:

How are magnesium alloys welded?

Gas, arc and spot welding are all efficiently practicable with Dowmetal Magnesium Alloys. The welding of magnesium alloys has been used in regular production of parts and assemblies for several years, a fact which has greatly broadened the usefulness, in many industrial fields, of this valuable weight-saving metal—for magnesium is the lightest of all structural metals.



Any good welder can join magnesium alloys as readily as he can other metals. *Gas welding* is well adapted to the making of butt welds; fluxes are available for this type of weld. In *arc welding* (illustrated at the right) a tungsten electrode is used, and the weld area is shielded from air by a screen of inert gas, such as helium or argon, thus eliminating the need of fluxes. Lap, butt, edge and fillet welds can be made by this method. Magnesium alloys can readily be *spot welded* to each other. Less current density is required because of the lower electrical conductivity of magnesium alloys in comparison with aluminum. Clean metal and clean electrodes will insure a better spot weld.

The comprehensive facilities of Dow's own fabrication shops, supplemented by the full experience and specialized knowledge which naturally accrue to the pioneer and leading producer of magnesium, are at your disposal. Inquiries regarding the welding of Dowmetal—to be done in your plant or by Dow—will receive prompt and thorough attention.

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where $V_{a_4b_4}$ is the relative velocity between points A and B of link 4.

The accompanying figure shows the location of a_4 and b_4 in the velocity polygon. Note that $a_4b_4 = ab \sin \theta$.

Then the Coriolis component may be expressed in terms of distances scaled from the velocity polygon as follows:

$$2\frac{a_4b_4}{AB}dc = 2\frac{ab}{AB}dc \sin \theta$$

Inspection will show that this does not equal $2(ab/AB)ec$, but $2(ab/AB)e'c$ where $e'c$ is perpendicular to ab extended, as shown in the accompanying figure.

In our article an attempt was made to present a general method of attack rather than the best possible solution. Although engineers wish to use the best possible solution for a particular problem, the possession of a general method of wide application is of more value than the knowledge of a special solution such as that applied in the author's *Fig.* 3. The circle of inflection method is, of course, a general one.

Mr. Beskin has rendered a service by presenting these solutions to your readers.

—A. S. Hall
—E. S. Ault

We wish to thank Messrs. Hall and Ault for also pointing out a typographical error in the article. The denominator in Equation 4 should be O_2A_2.—Ed.

To the Editor:

The author is indebted to Messrs. Ault and Hall for correcting the error in *Fig.* 3 of his article. Line *ec* should, of course, have been drawn perpendicular to *ab*, and not to *cd*, as shown in *Fig.* 3.

With respect to the comments concerning general methods and special solutions, it appears to the writer more general to use an auxiliary point which coincides with the point to be examined rather than a couple of two arbitrary intermediate points.

—Leon Beskin

". . . are the most complete"

To the Editor:

Referring to your current series of "Materials Work Sheets", published monthly in Machine Design, these cover everything we require and more. I think they are the most complete I have ever come in contact with.

—Kurt Ziehm
Felt & Tarrant Mfg. Co.

Reprints of the Materials Work Sheets published thus far will be made available shortly under one cover, at nominal cost.—Ed.

EACH BOEING FLYING FORTRESS has 3½ miles of copper wire for purely electrical purposes, 73 electric motors, 134 light bulbs and 300 radio tubes.

Selecting Steel on Basis of Hardenability

(*Continued from Page* 134)

necessary to obtain this maximum hardness depends on the alloy content of the steel, and therefore on its hardenability. Ordinarily full hardening is considered accomplished when a hardness of five to seven rockwell C below the maximum possible is obtained. For plain carbon steels this hardness would be five points still lower as depicted in *Fig.* 7. Ordinarily it is considered that lower hardness than full hardening, as just defined, will yield properties, when tempered, that are equal to those resulting from tempering after full hardening.

Determining Acceptable Hardness

If we draw a curve in *Fig.* 13 at a distance of ten points rockwell C below the curve showing maximum hardness obtainable, we will show the hardness for each carbon content that may be considered as the acceptable hardness in the quenched condition for obtaining suitable properties after tempering. *Fig.* 13 shows that 50 rockwell C is acceptable as full hardening for a steel containing .40 per cent carbon. If, therefore, we wish our 2-inch diameter shaft to harden to 50 rockwell C we should use a .40 per cent carbon steel with enough hardenability so that 50 rockwell C is obtained at a cooling rate of 11 degrees per second (the cooling rate at the center of a 2-inch round quenched in oil). Such a steel would have to harden above 50 rockwell C to a distance of 12/16-inch from the end of a Jominy end-quench hardenability bar. The hardenability curve of such a steel would be as shown in *Fig.* 12. This curve is typical of a number of steels such as 3140, 3240, 4140, 4340, 8740, and so on, but all heats within the chemical specification limits of these steels would not have the required hardenability to fulfill the requirement of 50 rockwell C at the center of a 2-inch diameter shaft, oil-quenched.

Fig. 14 shows how 8740 may vary in hardenability as the chemical composition varies within permissible limits. Hardenability is low when the alloy content is at the low specification limit and highest when the composition is on the high side of specification. Only a few steels have composition limits that would always insure a minimum of 50 rockwell C at the center of a 2-inch diameter shaft quenched in oil. Such a steel is 9540, having a minimum hardness of 50 rockwell C at about 12/16-inch on the Jominy bar. Another steel that will meet this requirement is modified 9262 containing .25 per cent molybdenum.

There is a great deal of 4340, 3240, 4140, 8740, and steels of similar hardenability used in highly stressed members of 2 inches or more in diameter. The obvious conclusion is that full hardening at the center is not necessary. Whatever hardness is desired in the as-quenched condition, *Figs.* 8, 9 and 10 can be used to determine the hardenability limits for the steel that will meet the requirements.

(*Concluded in next issue*)

Stainless Steel Lined Polymerization Reactors in Synthetic Rubber Plant

Equipment of Stainless Steel, Nickel and Monel meets many specialized requirements

Chemical engineers have met America's wartime challenge.

They opened the gates to a mighty flood of products going to war...strategic raw materials, synthetic substitutes, and entirely new substances having advantages all their own.

A factor in this production success is the wide use of stainless steel, Monel, and other corrosion-resistant alloys containing Nickel.

For in the chemical industry corrosion is a large-scale menace.

To wage war on this enemy, chemical engineers enlisted the aid of Nickel, because Nickel imparts to other metals strength and resistance to corrosion and wear. In the chemical field, as in many others, a little Nickel goes a long way to keep equipment producing.

It prolongs the life of processing apparatus, and protects the purity, color, and uniformity of the product.

Hence, stainless and Nickel alloys are specified widely for acid heaters and caustic coolers, for high-pressure autoclaves and vacuum evaporators, for cracking towers and polymerization reactors, for shipping drums and tank cars, for pumps, piping and storage tanks, for agitators and settlers, for stills and digestors—for every type of equipment that converts laboratory experiments into full-scale chemical operations.

For years we have enjoyed the privilege of cooperating with technical men of the chemical industry...and of many others. Whatever your industry may be ...if you want help in the selection, fabrication, and heat treatment of alloys... we offer you counsel and data.

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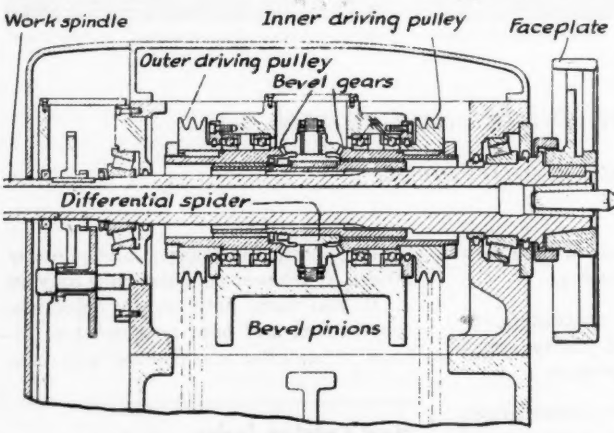
THE INTERNATIONAL NICKEL COMPANY, INC., *67 Wall St., New York 5, N. Y.*

Noteworthy Patents

Differential Infinitely Variable-Speed Drive

WHEN it is desired to provide a very wide range to control a variable-speed electric motor, a combination of both armature and field regulation customarily is used in conjunction with a shunt-wound adjustable-speed motor. However, for all speeds below the motor's basic speed, which is the normal low speed at which it is designed to operate, the motor is in the speed range where regulation is poor, the speed varying according to the load as well as to the speed setting. Such an arrangement, by itself, is totally inadequate to meet the demands found in a lathe headstock application in which heavy cutting is invariably done at relatively low speeds requiring maximum horsepower in these low speed ranges. A design which provides the desirable features of stepless speed through the entire range and which also is capable of producing sufficient power at the lower speeds is covered by patent 2,347,259 recently assigned to The R. K. Le Blond Machine Tool Co.

Basic principle of the transmission involves the provision of a pair of driving motors each connected through a differential transmission mechanism to a work spindle. Speed of each motor can be adjusted through the use



Two adjustable-speed motors belted to outer and inner driving pulleys actuate work spindle through differential transmission mechanism, providing infinite range of speed from zero up to synchronous speed of the motors

of conventional control mechanisms down to but not below the basic speed. The differential mechanism is shown in the accompanying illustration. The two driving pulleys are belt driven from the two separate motors. When both run in the same direction, the speed of the differential spider which is keyed to the work spindle is equal to half the sum of the two pulley speeds. When the direction of one pulley is reversed the work-spindle is then equal to half the difference between the pulley speeds.

Assuming that each of the two motors has a three-to-one speed range, top speed being 1800 and basic speed 600 revolutions per minute, all speeds from 1800 down to 600 are obtained by controlling simultaneously the speed of both motors. For lower speeds one of the motors is reversed, making it possible to operate the work spindle at practically zero revolutions per minute while maintaining at least basic speed on both motors.

Provision is made for selecting all speed ranges throughout the entire range by means of a single dial controlling appropriate rheostat mechanism and including reversal of one of the motors together with acceleration of the other at the appropriate speed setting. By properly proportioning the speeds of the two motors the spindle may be reversed at low speed, while reversal of both motors permits the spindle to be operated at high speed in the reverse direction.

Measuring Shaft Horsepower

DIRECT indication of horsepower output of a machine utilizing wholly mechanical means is possible through the use of a design covered by patent 2,345,444, recently assigned to Square D Co. Two movements—one proportional to torque and the other to speed—are hooked into a linkage which produces a movement proportional to their product.

In the accompanying illustration the mechanism is shown as applied to an aircraft engine employing hydraulic cylinders to balance and indicate the torque reaction on the stationary sun gear of the planetary transmission. A movement proportional to the torque developed by the engine is therefore available at the power meter through the use of a Bourdon tube subjected to the same pressure that exists in the torque reaction cylinders.

Speed indication is derived from a flyweight type of tachometer driven at a speed proportional to that of the engine. In order to obtain a displacement which is a linear function of the engine speed, the collar is constrained by means of a flat spring whose movement is limited by adjustable stops, permitting the movement-versus-speed characteristic to be varied within wide limits. Movement of the collar is transmitted to a motion-amplifying mechanism for actuating the pointer of the power meter. The mechanism includes a bell-crank lever pivoted as shown and engaging, with one forked end, the groove of the collar. The motion-amplifying mechanism also includes a device for varying its ratio of transmission, consisting of a rod mounted in a frame for movement about a vertical pivotal axis. The rod is moved by means of a pin on the frame engaged by the free end of the bell-crank lever. A ratio slider is provided for connecting the rod and a lever attached to the toothed segment which

Double Row PN Needle Bearing Gives Greater Stability

ESPECIALLY IMPORTANT UNDER COUPLE-LOAD CONDITIONS OF AIRCRAFT PULLEY OPERATION

•

The double row PN Type Torrington Needle Bearing has unusual stability as well as a low friction coefficient under widely varying degrees of cable pull-off or misalignment.

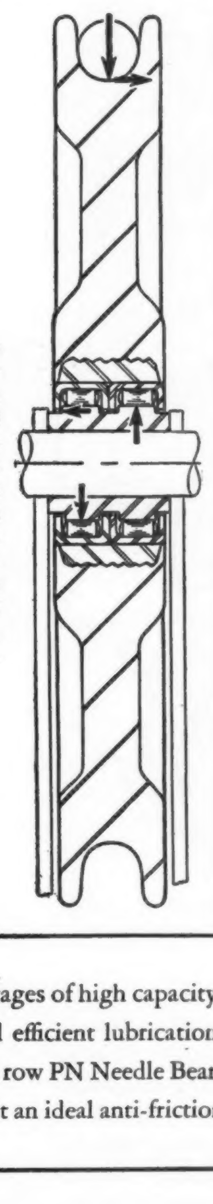
The double row PN Needle Bearing is, therefore, ideally suited to such applications as aircraft pulley operation where perfect alignment is difficult to obtain and where friction must be held to a minimum under all operating conditions.

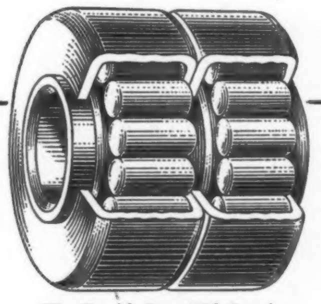
The accompanying cross-section indicates how the use of two rows of small diameter needle rollers resists the couple-loading encountered—one row carrying the couple in one direction; the other resisting the couple in the opposite direction. Thus the exterior force is resisted by the radial forces within the bearing which are properly distributed in opposite directions on the two sets of needle rollers.

Under conditions of severe misalignment and heavy loads, tests have conclusively demonstrated that Needle Bearing equipped aircraft pulleys will maintain pulley alignment with minimum friction.

This increase in stability coupled with other advantages of high capacity, light weight, and efficient lubrication, which the double row PN Needle Bearing offers makes it an ideal anti-friction unit for aircraft pulley and similar applications.

Cross-section showing Double Row Needle Bearing equipped pulley under misalignment. Arrows indicate direction and distribution of the couple loading in opposite directions on the two rows of anti-friction needle rollers.





The Double Row PN Bearing

Needle Bearing equipped aircraft pulleys* are currently available from leading pulley manufacturers who will gladly supply further information.

Test data on operating characteristics of the PN Needle Bearing under many types of service conditions is available. A copy of this data for your engineering files will be supplied on request.

**Fully conforms to AN-FF-P-796 specifications for aircraft pulley bearings.*

• • •

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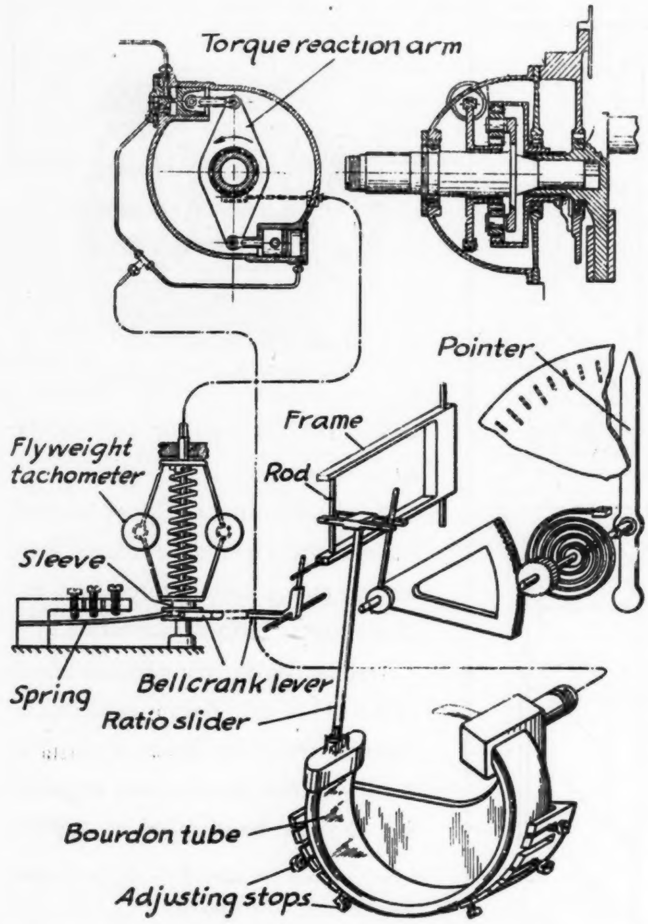
TORRINGTON NEEDLE BEARINGS

actuates the pointer mechanism. An increase in engine speed will cause an upward movement of the collar, resulting in lateral displacement of the rod through the bell-crank lever. The displacement of the rod causes a clockwise rotation of the toothed sector about its axis and a counterclockwise movement of the pointer relative to the dial.

The pointer movement is multiplied in response to the engine torque as a result of movement of the ratio slider.



Speed and torque indications are combined in power meter to give an indication of horsepower output

As indicated in the figure, an increase in reaction on the sun gear results in a corresponding increase in pressure in the hydraulic cylinders and in the Bourdon tube to which they are connected. The corresponding movement of the Bourdon tube causes a downward movement of the ratio slider, increasing the ratio of transmission of the motion-amplifying mechanism. The pointer is thus jointly operated by the torque and speed-responsive means, causing a total deflection of the pointer which becomes a measure of the engine's power. The dial may accordingly be graduated in horsepower units. Since a displacement of the ratio slider at a short distance from the pivotal axis will cause a considerably greater change in the ratio of transmission than an equal displacement at a greater distance from the axis, the movement of the Bourdon tube may be correspondingly restrained at high pressures by means of the adjusting stops shown.

Inasmuch as power is the product of torque and speed, the torque-responsive and speed-responsive units may be transposed if desired, using the Bourdon tube to actuate the pointer while the tachometer changes the ratio. The power-meter mechanism can, of course, be adapted to indicate any function which is the product of two quantities that can be measured.

A Plan for Postwar Germany

RECOMMENDING selective restriction and control of German industry rather than indiscriminate destruction, the presidents of five national engineering societies —A.S.C.E., A.I.M.E., A.S.M.E., A.I.E.E. and A.I.Ch.E.— recently issued a joint statement outlining four steps, any one of which would prevent that country from waging or preparing for another war. These steps are:

1. Eliminate all synthetic oil capacity and prohibit the reconstruction of plants and the importation of oil beyond normal peace-time inventories. This would destroy the major part of Germany's internal oil resources. Coal, the raw material for synthetic oil, is plentiful in Germany and only a small percentage is used in synthetic oil plants. It is not readily controllable in the Reich.

2. Eliminate 75 per cent of Germany's synthetic nitrogen plant capacity and prohibit reconstruction of plants and all importation of nitrogen compounds. This will leave a capacity in Germany ample for peace-time nitrogen requirements but insufficient for production of explosives on a large scale. The relatively small amount of dynamite required for mining, quarrying, etc., should be under import control.

3. Eliminate 50 per cent of Germany's steel-making capacity in those categories of plants which are most capable of producing essential war materials such as heavy forging, electrolytic and high alloy steels. Manganese, chromium, nickel and tungsten are practically nonexistent in Germany. Also prohibit importation of iron ore, flux material, steel and steel products beyond normal peace-time inventories.

4. Eliminate aircraft plants and equipment. Aluminum and magnesium are the raw materials required for airplane manufacture. There are no important bauxite deposits in Germany and importation should be prohibited. Aluminum and aluminum plants should be destroyed and importation of aluminum ingots beyond prewar peace-time needs be prohibited.

By attacking the problem from this angle, it would be possible to set up uncomplicated, nonpolitical controls to prevent the rearmament of Germany, but at the same time make it possible for the German nation to meet its own peace-time needs and to help in the vast task of restitution and reconstruction. It cannot be overlooked, however, that any program to take the military "sting" out of Germany will require supervision and vigilance for a long period in the future.

Engineers play an essential part in providing employment in the economy of any nation. Over here, especially, they see their function not in narrow, professional terms, but in providing jobs and in promoting industrial production. They do not believe that crippling the normal peace-time industrial economy of any country, even an enemy nation, can promote world peace and reconstruction. On the contrary, such a policy jeopardizes the peace and progress of all.

Worn or Broken Surfaces can be BUILT UP *quickly* with "997" Low-Fuming Rod

With five teeth gone from this large gear and an important production unit stopped dead in its tracks, what's a Production Manager to do? A new gear meant patterns, castings, days of machining—weeks of delay. Pegging the broken teeth would cut down the delay—but increase the hazard of a repeated failure.

It's in a pinch like this that the importance of Bronze Repair Welding with the oxy-acetylene torch is most appreciated—for by using this fast, economical method of repair the gear was welded, machined and back in operation in a few days. Low temperature bronze welding with "997" Low-Fuming, Tobin Bronze* and other Anaconda Welding Rods is so controllable that it is often possible to weld parts without dismantling, and in many instances, finish by filing to a template.

"Don't Scrap It...Bronze Weld It!" is more than a slogan—it's a NECESSITY in many plants. Publication B-13 contains suggested welding procedures and examples of savings in repair time and expense. Write for a copy.

*Reg. U. S. Pat. Off.

Let's help the other fellow, too.

Under the stress of present industrial output, mechanical production equipment often feels the strain. Bronze Welding is being used extensively, not only for the repair of iron, steel and copper alloy parts, but also for the building-up of worn surfaces. If you've done an interesting bronze welding job that you feel would help the other fellow, tell us about it. If of general interest, we'll pass the word along in these pages.

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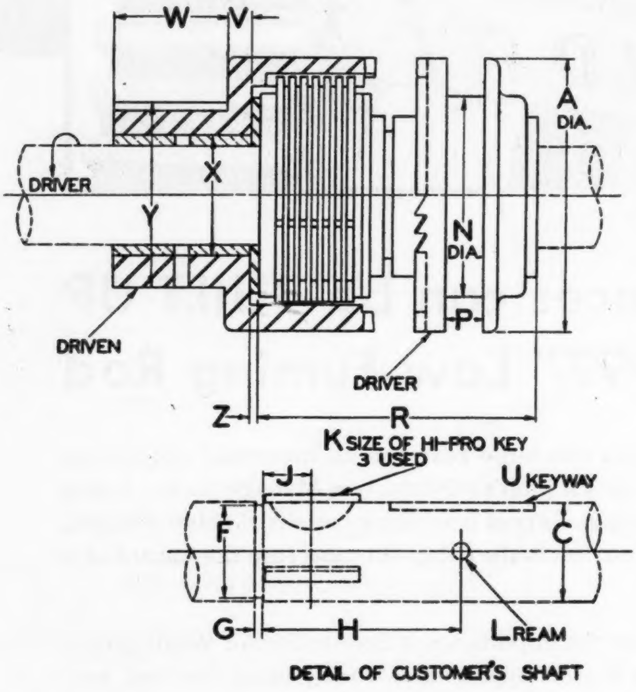
Anaconda Bronze Welding Rods

New PARTS AND MATERIALS

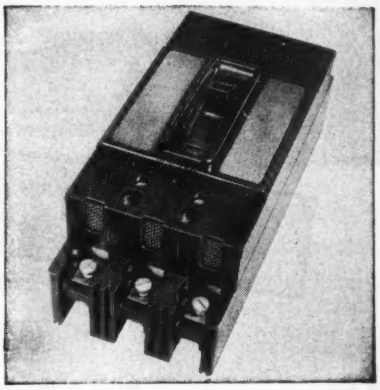
Multiple-Disk Clutches

COMBINATION multiple-disk and jaw-drive clutches, announced by Rockford Drilling Machine Division, Borg-Warner Corp., Rockford, Ill., are valuable in high-speed drives requiring clutch action for starting, followed by shockless positive drive. The clutches are designed particularly for use in machines handling thread, yarn, paper or other relatively fragile materials. First movement of the shipper sleeve brings the driven member up to speed with the smooth action that makes the clutches applicable in a variety of automatic and semiautomatic machinery. Further movement of the shipper slides the teeth of the jaw clutch into shockless engagement, under no-load conditions, for positive continuous driving. Pressure is then released on the disks. Capacities of the new clutches range from 2 to 25 horsepower at 500 revolutions per minute.



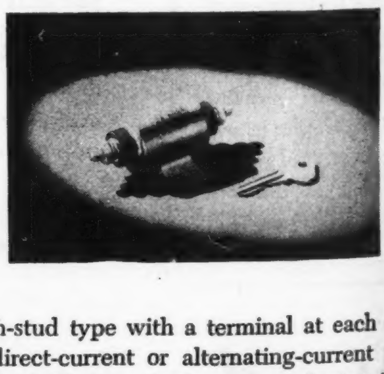
100-Ampere Circuit Breaker

KNOWN AS THE De-ion circuit breaker, the new 100-ampere unit announced by Westinghouse Electric & Mfg. Co., Pittsburgh, requires little space and permits light structures for distribution panelboards, built-in specifications and bus duct plug-ins. All ratings are available in the breaker with uniform pole spacings and terminal arrangement, providing interchangeability between ratings. The F-frame permits a 100-ampere, 600-volt alternating-current or 250-volt direct-current breaker in the same space as required by the 50-ampere, 600-volt alternating-current or 250-volt direct-current rating. The fuseless breaker is equipped with thermal and instantaneous magnetic trip elements, and provides maximum loading of circuits and quick resumption of interrupted service. Contact pressure increases with wear, prolonging life of the silver alloy contacts and breaker. Both two and three-pole units are available.



Noise Suppression Capacitor

DESIGNED SPECIALLY to reduce radio-noise voltage from generators, inverters, motors and other equipment, a new Pyranol radio-noise-suppression capacitor has been announced by General Electric Co. These units are of the through-stud type with a terminal at each end. One line of a direct-current or alternating-current power circuit can be "fed" through the unit, reducing internal inductance and resistance, and increasing filter efficiency



for a given capacity. Compact and light in weight, the capacitors are especially effective in reducing radio noise at higher frequencies. Physical dimensions are 1¾ x 3⅝ inches, and the unit weighs 4½ ounces. The capacitors can be mounted in any position and will operate over a temperature range of 50 to −50 degrees Cent. They are rated at 0-100 amperes, 250 volts maximum alternating or direct current, .55 microfarad, and are designed to meet vibration tests required by the U. S. Army Air Forces Specifications.

Accumulators for Hydraulic Systems

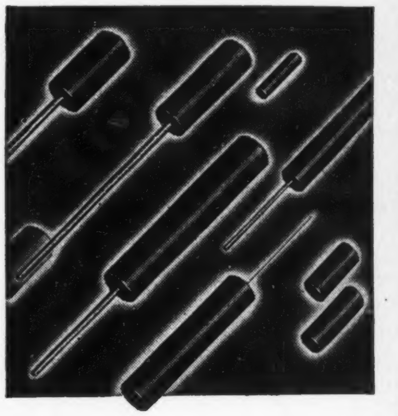
DEVELOPED FOR aircraft, new hydraulic accumulators are now being offered by Greer Products Corp., 39 West Sixtieth street, New York 23, for application to machine tools, automotive, railroad and marine equipment. The new accumulator consists of a one-piece chrome-molybdenum steel shell which contains an enclosed one-piece synthetic rubber bladder having an integrally molded air valve. The steel shell is heat treated so that all internal stresses are relieved, permitting sufficient elongation for shock loading. Molded of synthetic rubber which is impervious to hydraulic fluids, the bladder is capable of operating at temperatures ranging from —67 to 180 degrees Fahr. Its tensile strength is sufficient to withstand 200 per cent elongation without permanent set. The units are available for operating pressures up to 3000 pounds per square inch, in capacities of .333, 1, 2½, 5, 10 and 25 gallons. For larger capacities a number of accumulators may be used. In operation the accumulator is preloaded to the required pressure. After the initial charge, the unit needs to be recharged only every three or four months, the hydraulic fluid being introduced under pressure through a port in the end of the shell. When the hydraulic system is in need of the reserve fluid stored in the accumulator, the bladder expands until the fluid is forced completely out of the shell.

Buoyant Plastic Foam Material

IN ADDITION TO war uses of the new plastic foam recently introduced by United States Rubber Co., Rockefeller Center, New York, peacetime applications that are foreseen include insulation for trains, airplanes and automobiles. The foam has great buoyancy with minimum weight, and is semirigid. Because of the air space the material has good insulation and sound-deadening properties in comparison to its weight which is less than a pound and a half per cubic foot. It can be made to weigh three-quarters of a pound per cubic foot also. In order to produce new and different war material, a combination of synthetic plastic materials are foamed and then solidified. The material is known as Flotofoam.

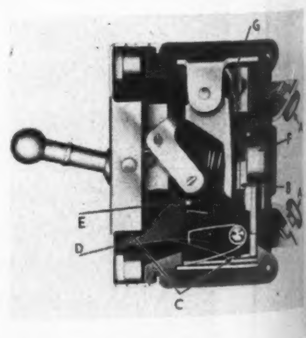
Side-Molded Iron Cores

RECENTLY INTRODUCED by Stackpole Carbon Co., St. Marys, Pa., a line of iron cores molded by means of pressure applied from the sides rather than from the ends has proved to have advantages for permeability tuning applications at broadcast band frequencies. Similar cores



now are available for short-wave frequencies including television and frequency modulation. In these side-molded cores, any density resulting from molding pressure extends evenly over the entire length of the core, assuring uniform permeability with respect to length.

Temperature-Resistant Breaker



EXTREME HIGH AND low temperature resistance of the new switch breaker announced by Littelfuse Inc., 200 Ong street, El Monte, Calif., and 4757 Ravenswood avenue, Chicago 40, is accomplished by bimetal which is used as the finger trigger. No appreciable mechanical load is exerted on the bimetal as it trips the breaker. The switch can take −60 to 350 degrees Fahr. without breaking. Primarily designed for military uses—aircraft, tanks, ships, landing craft, etc.—the control has a high time lag offering protection to motors and other equipment having high starting surge currents. Range of the breaker is 5 to 50 amperes at 32 volts, alternating or direct current. It is capable of breaking 2500 amperes on short circuit, and meets requirements of holding for one hour at 115 per

cent of rated current, breaks within the hour on 138 per cent of rated current, and breaks at 200 per cent of its load between 10 and 100 seconds. These tests were performed at the ambient temperature of 77 degrees Fahr. ±1.8 degrees Fahr. This switch-type, nontrip-free No. 1560 breaker is enclosed in a moistureproof black bakelite case, and is panel-mounted by two 6/32 screws, ¼-inch long, for 1/16-inch thickness of panels equipped with heavy copper terminal bus bars. Overall size is 2⅛ x 2 x ¾ inches.

Protective Coating

DEVELOPED BY Dow Chemical Co., Midland, Mich., a hot melt dip type of packaging known as Stripcoat protects, preserves and packages metal parts in one operation. The coating is easily removed by slitting and the metal part is ready for use, having been protected against cor-

rosion, abrasion and mechanical damage. Metal parts such as axle shafts, complete axle assemblies, cam shafts, connecting rods, gears and intake and exhaust valves can be dipped in Stripcoat. Numerous parts having internal surfaces can also be coated. If a vertical dip is used the coating will bridge the opening, preventing melt from traveling to the inner surface. Parts with blind holes of not more than one inch, and gaskets and other units where leather or felt is part of the construction can also be coated. An advantage of Stripcoat is that it sets quickly without the aid of mechanical drying equipment. According to the manufacturer, a saving of time of from 60 to 95 per cent has been reported, depending on the type of part being packaged. Resiliency of the material which allows for contraction and expansion of metal when subjected to temperature fluctuations is another factor.

Plasticized Fabric Offered

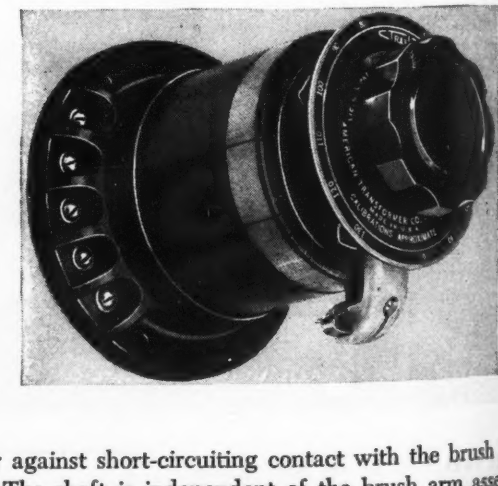
RECOMMENDED FOR numerous industrial applications where an abrasion-resistant and durable fluidproof, heat-resistant, heavy-duty fabric is desired, Cottonleather is being offered by the Southern Friction Materials Co., Charlotte, N. C., for covering rollers, pulleys, and treads in conveyor systems, chute linings, oil-soaked clutch facings, bearing wicking, textile frictions, etc. This plasticized fabric is available in standard sheets of 18 x 72 inches, while cut sheets are furnished in squares, rectangles and circles. The standard thicknesses range from 1/24 to 9/48-inch.

Electrode for Aircraft Welding

AN IMPROVED version of the smaller sizes of Airco No. 90 electrode has been announced by Air Reduction Sales Co., 60 East Forty-second street, New York 17. Known as No. 90-A, this electrode is especially designed for welding light-gage chrome-moly and similar steels used in aircraft welding. Made to a new coating formula, it provides smooth arc operation, good appearance of deposits, strong arc action, reduced slag interference, and operation at high currents without deterioration of coating at stub end. Available in all three of the most popular sizes for light-gage welding—1/16, 5/64 and 3/32-inch diameters—it is for both alternating and direct-current operation, straight or reversed polarity. Meeting the requirements of U. S. Army Air Corps Specification 10286-B, Type 1, Grade 2E, and the A.W.S. and A.S.T.M. Specifications for Classification E 6013, the electrode in tests, welded on SAE 4130, shows transverse tensile strengths of 80,000 to 90,000 pounds per square inch.

A-C Voltage Regulator

OFFERED BY American Transformer Co., 158 Emmet street, Newark 5, N. J., the latest Model TH Transtat alternating-current voltage regulator incorporates a number of developments. The brush arm of the regulator is of unique design, being an accurately machined die casting which permits good heat dissipation and protects the commu-



tator against short-circuiting contact with the brush holder. The shaft is independent of the brush arm assembly and can be removed by drawing one pin. By the use of a phenolic thermosetting plastic base, high dimensional conformance is assured and lead shorting is prevented. Other refinements include the vinyl acetal insulated wire, impregnation of core and coil with a synthetic phenolic resin varnish of the polymerizing type followed by baking, corrosion-resistant fittings, and a new dual mounting arrangement for open delta three-phase control. For 115 volts, 50-60 cycle input, it regulates the voltage from 0

GIVE W·A·B CONTROLS A PLACE IN YOUR PLANNING

"Donkeys" aren't stubborn *with* W.A.B. Controls

Paul Bunyan's greatest legendary exploits in juggling mammoth timber are duplicated daily by Donkey engines like this. It's a "never-sweat" job for the operator, for through the magic of air he governs the most complex operations by the movement of two small handles on the W-A-B Controls.

W-A-B devices are handling some of today's most responsible remote control jobs. An entire cycle of operations can be governed merely by positioning small handles. Operator fatigue is lessened, for there is no muscling of heavy levers, whether the operating forces are ounces or tons. Equipment damage is lessened, for interlocks prevent the setting up of opposing actions, and the sequence of operation cannot be accidentally varied.

If your post-war products or production involves a remote control problem, you'll find W-A-B Controls a proven answer. Give them a place in your planning.

Westinghouse Air Brake Company

INDUSTRIAL DIVISION

General Offices: Wilmerding, Pa.

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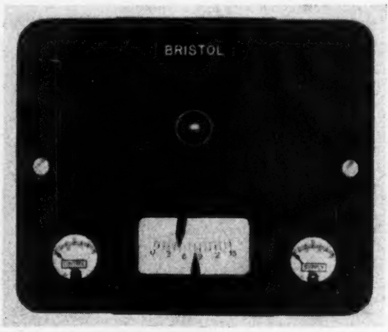
remote control systems

to 130 volts output. The power the regulator will deliver ranges from 250 to 590 depending on the type of regulator employed.

Air-Operated Controllers

INDICATING air-operated control instruments, known as Model 93 series, announced by The Bristol Co., Waterbury 91, Conn., have been developed for controlling temperature, pressure, vacuum, liquid level and humidity. Operating on the free-vane principle of automatic control, the units have a throttling range of from ½ to 15 per cent with the adjusting mechanism arranged so that changes can easily be made. The controllers are direct set instruments for controlling at any value within the range,



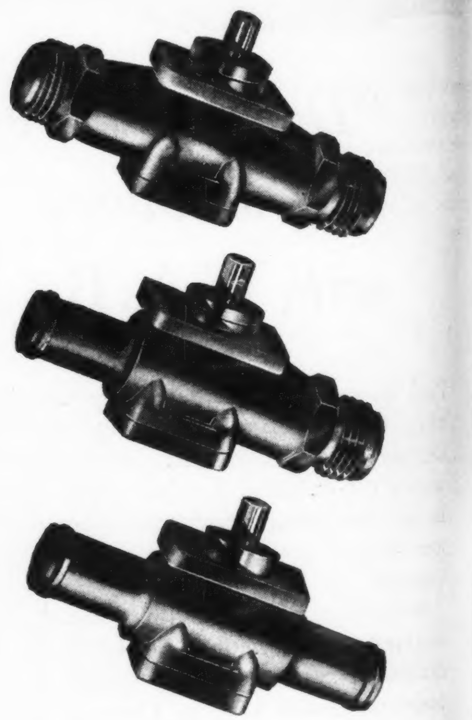
by turning the control pointer to the desired value, and can be changed by a finger adjustment from reverse to direct action or vice versa. Applications of the controller include bake ovens, drying ovens, cookers, retorts, plating tanks, and soft metal pots, and the control of steam pressure, pressure in retorts, and back pressure and digesters.

Self-Priming Rotary Pumps

LOW-PRESSURE, rotary pumps for all types of liquids having lubricating qualities have been developed by John S. Barnes Corp., Rockford, Ill. For use in oil pressure systems on automotive, truck or tractor equipment, it is also suitable for torque converters. Capacity ranges proportionately from one gallon per minute at 600 revolutions per minute, to four gallons at 2400 revolutions. It has a high volumetric efficiency for pumping low viscosity fluids. An outstanding feature of the pump is the spur-gear tooth form which eliminates excessive sliding and reduces slippage of the fluid to a minimum. Each tooth fills the mating space as the gears mesh, providing effective sealing. Displacement of fluid is thus assured despite variation in viscosity or other factors. A relief valve, optional on the pump, provides protection against high pressures. The vacuum created by two spur gears draws the fluid being pumped through the pump inlet. Passing between the gear teeth to the discharge side of the pump, the fluid is forced into the discharge line by meshing of gears. Driving gear is equipped with a free-floating type driveshaft with shear pin to minimize damage caused by foreign materials entering the pump housing. A spring-loaded oil seal is used on the driveshaft and the shaft is tanged to fit the power drive slot. The spur gears are alloyed cast iron, cast integral with the shaft itself. All moving parts of this self-priming pump are self-lubricated.

Aircraft Fuel Line Valves

CONTAINING THE latest improvements in design for valves used in conjunction with highly volatile liquids, three new types of aircraft fuel line valves for aromatics are being manufactured by the aircraft parts division of Davidson Mfg. Co., Los Angeles. The danger of cracking the valve body during assembly because of overtightening the tube fittings has been eliminated by making the AN thread integral with the body, obviating need for

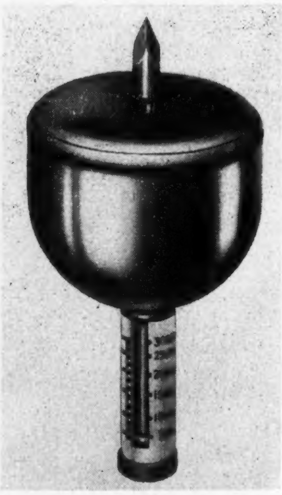


special fittings. Elimination also of unnecessary parts has made the unit lighter without loss in strength or performance. Under United States Army winterization tests, the valves will operate under any temperature conditions ranging from as low as −65 degrees Fahr. to above fuel boiling points. The valves have proved leak and stick-proof. Maximum initial breakout torque ranges between 12 and 18 inch-pounds but only 6 inch-pounds torque is required to move the valve to open or closed position after breakout. It operates on a 90-degree turn to either on


WAR TESTED ON AIRCRAFT ALL OVER THE WORLD
NOW AVAILABLE FOR INDUSTRIAL APPLICATIONS
3 PIECES (EACH REPLACEABLE)
ASSEMBLY WITHOUT SPECIAL TOOLS. NO TIGHTENING OR ADJUSTMENT AFTER ASSEMBLY. FITTINGS CAN BE REMOVED FROM HOSE AND RE-USED OVER 100 TIMES.
HOSE CUT TO LENGTH
NIPPLE
1
NUT
2
SOCKET
3
AEROQUIP CORPORATION
JACKSON, MICHIGAN, U. S. A.
LOW PRESSURE
1/2"
3/8"
1/4"
1/8"
MEDIUM PRESSURE
3/4"
5/8"
1/2"
3/8"
5/16"
1/4"
3/16"
HIGH PRESSURE
3/4"
5/8"
1/2"
3/8"
5/16"
1/4"
3/16"
LOW AND MEDIUM PRESSURE
1 1/2"
1 1/4"
1"
★ ★ ★ 303 WAREHAM BLDG., HAGERSTOWN, MD. - 1709 W. 8th., LOS ANGELES ★ ★ ★

or off positions with a spring detent at both to maintain placement. The valves need no lubrication and are furnished in the following three types with ends to customers' specifications: AN-816 threads at each end, or AN-816 threads at one end and 1-inch hose fitting at the other, or 1-inch hose fittings at both ends.

Tachometer Introduced



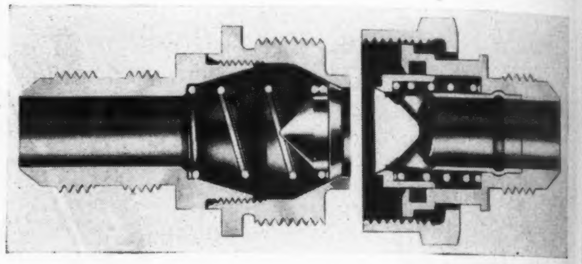
WEIGHING ONLY 5½ ounces and being of small size—2¼ inches in diameter—the new tachometer of The Standard Machinery Co., Providence, R. I. has a range from 500 to 3600 revolutions per minute. Recordings in revolutions per minute are easily read and without the use of timing or counting devices. Readings are constant and record fluctuations. The scale employs black figures against an orange background. A pointed contact spindle is a part of the instrument for use with shafts that are centered and an elastic tip is furnished that will slip over the pointed spindle for use on shaft ends that are not centered. The tachometer is moisture and dustproof, and has a baked enamel protective coating on all surfaces except the scale which is enclosed in a plastic tube.

7-Day, Dial Time Switch

EQUIPPED WITH 6-inch calendar dials which make one complete revolution every seven days, the new 700 Series switch of Paragon Electric Co., 707 Old Colony building, Chicago 5, can be used in timing automatic heat, ventilating, lighting, pumping or flushing operations. Dial trippers can be set independently for different daily on and off schedules, and settings can be made in advance for an entire week. Any day or days operations may be omitted entirely on a preset program. Each day of the week is clearly separated from other days, as well as day from night. Graduations of hours and half-hours are also provided. Operations from on to off or from off to on can be set to three-hour intervals.

Self-Sealing Coupling

CONFORMING WITH Specification AN-C-05, the improved self-sealing coupling of Aeroquip Corp., Jackson, Mich., permits disconnection and reconnection of hydraulic or other lines without loss of fluid or inclusion of air. The seal is affected by an O-ring embedded in recess of body. One half of the coupling can be provided with a detachable mounting flange. On connecting the coupling halves, the protruding portion of the one half makes contact with the sleeve of the other body, simultaneously expelling the air between the mating parts and preventing it from entering the system. Further movement of the union nut will move the halves until the position allowing



free passage of fluid is reached. In service it has been proved that there is practically no restriction when the coupling is completely coupled and consequently self-sealing couplings can be used in suction lines.

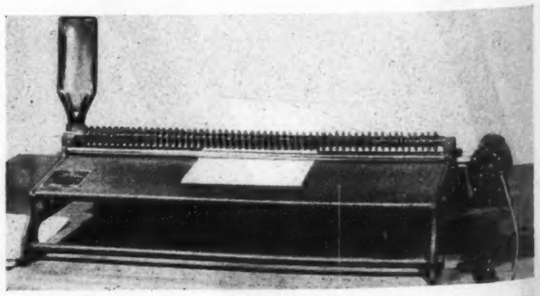
Engineering Dept. Equipment

Slide Rule, Decimal Locator

DETERMINING a decimal point mechanically in involved expressions with results up to 19 digits or 19 zeros is one of the functions of the new slide rule decimal point locator introduced recently by Pickett & Eckel, 53 West Jackson boulevard, Chicago 4. Another feature of the new rule is obtaining 30-inch scale accuracy for cube root and 2-inch scale accuracy for square root on a rule with 10-inch scales. One setting of the hairline enables the computer to read square root, cube root and logarithm, and one setting also determines the decimal point location for square root and cube root.

Continuous Printer, Developer

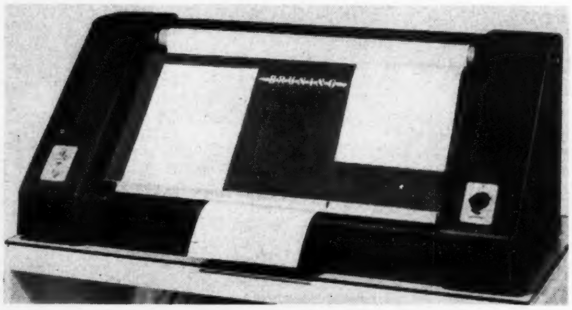
TWO NEW MACHINES have been announced by Charles Bruning Co., Chicago—a continuous printer and a continuous developer. The BW-Copyflex Model 2 printer makes possible the duplication of drawings as well as



typed, printed or illustrated matter. It exposes tracings, line drawings, specifications, Van Dyke negatives, blueprints, etc. Original material with copy on both sides can be reproduced on either side or both sides. Prints are developed in trays and dried in a simple drier, readily

available. By using a switch, this printer can be used for exposing black and white prints also. It fits in desk-top space and exposes roll stock or cut sheets up to 24 inches wide, at a speed of 5 inches to 30 inches per minute.

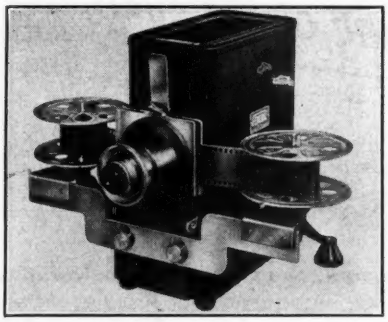
The other new machine—Model 153-M BW developer is used for developing black and white prints exposed on the



Model 2. Prints are delivered ready for use. This machine develops cut sheets or roll stock up to 24 inches at a speed of 12 feet per minute.

Microfilm Reader-Projector

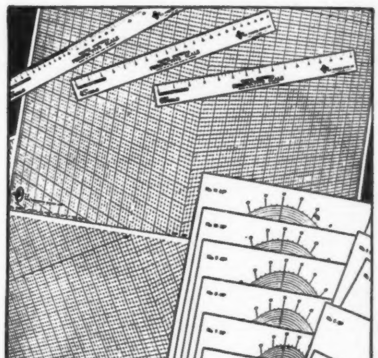
ORIGINALLY DESIGNED and built for the Armed Forces, the portable microfilm reader-projector has been made available to industry in general by Federal Mfg. & Engrg. Corp., 199-217 Steuben street, Brooklyn 5. In engineering departments where microfilming is used to reduce drawings to tiny rolls of film, the reader-projector offers an opportunity to study any drawing at any time. It will reproduce microfilmed letters, documents, records or drawings to full size or larger, on any convenient wall or screen. Weighing approximately 10 pounds, it oc-



cupies very little space. The 360-degree rotating projection head projects either vertical or horizontal images. Reel holder takes 100-foot reels of 35 millimeter, unperforated or perforated film, either black and white or colored, double or single frame. A short length of strip film can also be projected. For 16-millimeter film an adapter can be supplied. Magnification is from 5 times to over 24 times—five times about 15 inches from screen, and 24 times about 5 feet from the screen. A 100-watt projection lamp is standard equipment, but the reader can be used with 150 or 200 watt lamps. The size of the unit is 5¾ x 11 x 11⅝ inches.

Graph and Circle Instruments

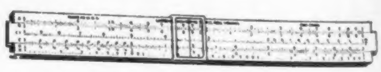
PERSPECTIVE DRAWING instruments recently developed by Charles W. Downs & Son Co., 2280-2300 Fourteenth street, Detroit 16, are now being offered to engineers and designers. The instruments reduce perspective art to simplest terms and make possible a degree of accuracy not achieved by any but the most skillful perspective artist. The draftsman can go further toward presenting his subject as it actually appears to the eye. Included in the Truper line are a wide variety of perspective graphs and circles. Perspective scales show diminishing units of measurement as they recede toward the vanishing point at the designated angles. Seventeen templates are included in a set, showing concentric circles .25 to 8.00-inch diameters in true perspective at every angle from 0 to 90 degrees in increments of 5 degrees. For ease in locating points on a circle, each template is divided into 5 degree segments. The Series B graphs are



furnished nine to a set, showing mutually perpendicular planes at angles of incidence from 5-85 to 85-5 degrees, with lines every ¼-inch, heavy lines every inch, and dimensioned in true perspective. The working area of each graph is 16 x 21 inches. The Series C graphs are similar to Series B, but are for use in drawing subjects below or above eye level. Seventeen graphs are in this set.

Slide Rule Available

OF STANDARD 10-inch length with white composition face, the new slide rule announced by Lawrence Engineering Service, Peru, Ind., has both red and black figures, with A, B, C, D, Cl and K scales on the front of the rule and the S, L and T scales on the back of the slide. The recess at the end is fitted with a hairlined, trans-



parent plastic lens, permitting the use of the S, L and T scales without reversing the slide rule. The slide indicator on the front is metal-framed and also has a hairlined, plastic lens. The wood, it is claimed, is impervious to climatic changes, and retains accuracy under adverse tropical conditions.

D. F. Newman

Christian Steenstrup

Russell H. Lasche

MEN... of... machines

CHRISTIAN STEENSTRUP, a well known engineer who developed the hermetically sealed refrigerator unit, has been relieved of his responsibilities as head of the refrigerating engineering division of General Electric Co., with which organization he has been connected for forty-three years. He will, however, continue as consultant for the present. An advocate of the modern mass production of electric refrigerators, Mr. Steenstrup has 113 patents covering many phases of refrigerator engineering and manufacture. Born in Denmark, he came to America at the age of 14, and in 1901 started to work for General Electric as a toolmaker. His inventive genius coming to the fore, he developed an automatic indexing device to eliminate the slow hazardous hand method then in use on the punch presses. Later he became assistant foreman of the department. During the first World War he was superintendent of the munitions department, and a few years hence was appointed supervisor of mechanical research, being directly responsible for the design of special equipment. In 1925 he produced the new hermetically sealed refrigerator unit suitable for mass production.

D F. NEWMAN, engineer succeeding Christian Steenstrup as head of the refrigerating engineering division of General Electric Co., was formerly assistant engineer. A native of Amsterdam, N. Y., Mr. Newman joined General Electric in 1907. Two years later he was transferred from his position as a clerk to the apprentice course to learn pattern making. After graduation in 1913, he entered the mechanical superintendent's department to design special machine tools. In 1922 he was placed in charge of this designing work, under the direction of Mr. Steenstrup, and in two years was assigned to special work in the refrigerating engineering division. Mr. Newman has 27 patents, some on hydrogen copper brazing and others on refrigeration.

RUSSELL H. LASCHE has recently been appointed director of engineering and research for the Fairchild Camera & Instrument Corp., New York, manufacturers of aviation instruments such as cameras, radio compasses, sextants, gun sights, and other devices. Mr. Lasche, a graduate of the University of Wisconsin engineering school, has been connected with the company's various activities

for fifteen years. Beginning his career in the sales end, he later worked for Fairchild Aerial Surveys out of Chicago. He also organized the aerial photography program for the Colombian government where he took the first photographs of the Orinoco River system headwaters, and made the first air-map of the Caribbean coast. His position prior to his appointment was that in charge of sales of the company to the war department.

♦

D E. BATESOLE has been elected vice president and chief engineer of Norma-Hoffmann Bearings Corp, Stamford, Conn. Born in Toledo, O., he was educated at Ohio State university. Prior to joining the Norma-Hoffmann corporation, he was employed in the designing department of Willys-Overland Corp., and later taught engineering drawing at Ohio State. During World War I he served in the Signal Corps and later in the Engineers' Corps. Connected with the Norma-Hoffmann engineering department since 1917, Mr. Batesole has devoted much of his time to bearing applications in many diversified fields. He holds numerous patents covering bearing designs as well as bearing mounting designs. In 1937 Mr. Batesole was appointed chief engineer, the position he held before his recent appointment.

♦

BENJAMIN R. NEWCOMB, formerly president of the Auditorium Air Conditioning Corp., of which he remains a director, has been elected president and general manager of the John Waldron Corp. Prior to his connection with Auditorium Air Conditioning he was chief engineer of the American Optical Co.

♦

ARTHUR NUTT has resigned as vice president of engineering of Wright Aeronautical Corp., Paterson, N. J. His plans have not been announced as yet.

♦

ROBERT L. HARTLEY, previously field engineer for Chandler-Evans Corp., South Meriden, Conn., has joined the engineering department of Lincoln Engineering Co., Pawtucket, R. I.

♦

EARL R. KLINGE has become research engineer for Continental Aviation & Engineering Corp., Detroit. Formerly he had been an associate professor in the department of mechanical engineering at Pennsylvania State College.

♦

HAROLD W. SCHAEFER as assistant manager of the newly-formed radio receiver division of Westinghouse Electric & Mfg. Co., will be in charge of the division's engineering and production activities. Mr. Schaefer has a background of more than eighteen years of radio and other household equipment engineering experience.

♦

JULE GORDON, formerly aircraft design engineer for General Airborne Transport Co., Los Angeles, has joined Norman E. Miller & Associates, same city, in a similar capacity.

♦

COL. AL BODIE has returned to the United States on inactive duty and is now director of postwar engineering and manufacturing for United Aircraft Products, Los Angeles.

♦

S. K. LEHMAN, previously chief engineer of the New York Division of Standard Aircraft Products Inc., has become a partner of Lehman & O'Connor to act with W. S. O'CONNOR as consulting, design and sales engineer.

♦

KARL K. PROBST has been made director of research for Reo Motors Inc., Lansing, Mich. He had been employed on the Works Administration Staff, Chrysler Corp., Highland Park, Mich.

♦

GEORGE H. HUFFERD has been appointed vice president in charge of engineering, Weatherhead Co., Cleveland.

♦

H ENRY G. TARTER, assistant chief engineer, has been named chief engineer of the aircraft carburetor engineering department of Bendix Products Div. A graduate of Oregon State College and the University of Pennsylvania in mechanical engineering, Mr. Tarter was employed as test engineer at the Philadelphia Aeronautical Engine Laboratory, Naval Aircraft Factory for six years prior to joining the Bendix corporation in 1935. He was field and development engineer in the aircraft carburetor department of Bendix from 1935 to 1940 when he was promoted to assistant chief engineer. Mr. Tarter has been actively associated in engineering and development of the company's injection carburetor.

♦

LEONARD S. HOBBS has recently been elected vice president of engineering of United Aircraft Corp., East Hartford, Conn., while WRIGHT A. PARKINS has been appointed engineering manager, Pratt & Whitney Aircraft Div. Both have been connected with United Aircraft for over 15 years. In 1935 Mr. Hobbs was named engineering manager of Pratt & Whitney and in 1942 was elected director of United Aircraft. Mr. Parkins was appointed assistant chief engineer in 1938.

VOLTAGE REGULATORS "VIBRATION PROOFED"

How can a Voltage Regulator be made to operate efficiently in spite of vibration?

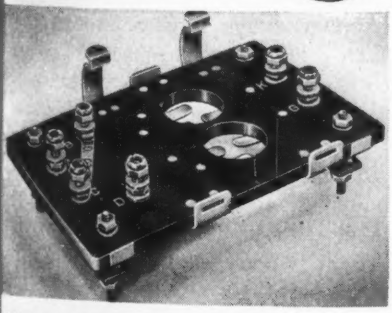
That's a question which faced design engineers responsible for the communications systems on our fighting planes. The answer was found in a little piece of scientifically compounded rubber safely bonded between two pieces of metal. When the regulators were mounted on these, the trouble was over.

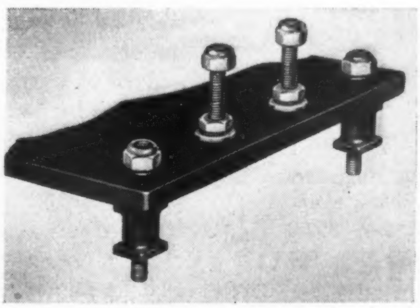
The mountings themselves look quite simple—but behind them is a backlog of scientific knowledge—the Science of Smoothness. By this is meant the science of engineering rubber and metal into resilient supports which will isolate vibration and shock.

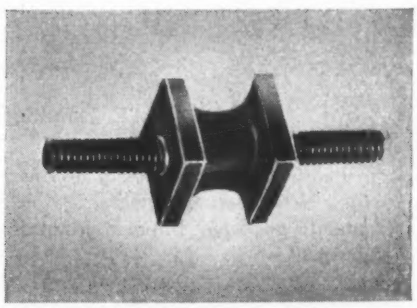
Men of the engineering staff of United States Rubber Company have been working in this particular field of science for many years. They have helped to solve a wide variety of difficult problems for engineers confronted with the necessity of preventing vibration and shock from interfering with the operation of their equipment.

They will be glad to work with you.

SERVING THROUGH SCIENCE







ONLY WHEN ISOLATED FROM VIBRATION can aircraft-type voltage regulators keep the flow of electricity constant for more efficient communication. This is accomplished through the use of "U.S." engineered rubber mountings designed specifically for each type of base. Quality of performance is predetermined before installation.

THIS VOLTAGE REGULATOR MOUNTING is only one of many types designed by "U.S." engineers for mechanisms ranging from sensitive electronic instruments to heavy industrial equipment.

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ASSETS to a BOOKCASE

High-Speed Combustion Engines

By P. M. Heldt, member of Society of Automotive Engineers; published by P. M. Heldt, Nyack, New York; 776 pages, 5¼ by 8 7/16 inches, clothbound; available through MACHINE DESIGN.

Written to serve both as a textbook for students and a reference book for engineers this latest edition of a familiar treatise seems to meet both requirements well. If one is concerned primarily with learning how the many units of a combustion engine operate, that story is here. If, on the other hand, one is interested in the technical aspects of the engineering and design involved, that story too is here.

At first glance the designer of, let us say packaging machines, may take the attitude that a book on combustion engines is of little if any significance to him. This is hardly the case, however, because the design of all mechanical devices is based on the same underlying principles of engineering. Thus, the designer of vacuum cleaners may find he can pick up many worthwhile ideas from the designer of sewing machines, the designer of cash registers may learn from typewriter designs, and so on.

Many phases of engineering are involved in the overall design of a combustion engine and these range all the way from stress analysis to thermodynamics. Since these engines are used so extensively in automobiles and aircraft, the quality of the engineering put into them is of the finest. Coverage of the book is complete, all of the many units such as cylinders, engine mountings, crankshafts and flywheels, ignition, bearings and lubrication, carburetors, etc., are discussed in an authoritative and comprehensive manner.

□ □ □

1944 SAE Handbook

Published by the Society of Automotive Engineers Inc., New York; 804 pages, 5⅜ by 8¼ inches, clothbound, available through MACHINE DESIGN, $5.00 *postpaid.*

Here is a reference work which should be in the hands of all design engineers. It makes available many quality standards, specifications and classifications pertaining to machine parts and materials which are the result of the combined efforts of some of our best engineering minds.

We need more standardization in engineering. Or perhaps it would be more accurate to say we need more general acceptance and widespread use of the many standards already available. There still is much too much time being wasted in many engineering departments on the design of units and parts which are readily available as standard commercial items.

In this latest edition a number of new features have been included. For example, there are the specifications for medium and heavy-duty coolant hoses; detailed standards for straight and tapered pipe threads; standards for power take-off and drawbar hitches, including safety protectors; standards for spring lock washers; and nomenclature for pistons and piston rings. In addition, much previously published data has been revised and brought up to date. These include classification of natural and synthetic rubber compounds; tables on steel hardness conversion numbers; standards for tube fittings on fuel and oil lines; and specifications for nonferrous metals including solders and both cast and wrought aluminum, magnesium, brasses, bronzes, and bearing and bushing alloys.

□ □ □

Light, Vision and Seeing

By Matthew Luckiesh, director of lighting research laboratory, General Electric Co., Nela Park, Cleveland; published by D. Van Nostrand Co. Inc., New York; 323 pages, 5⅜ by 8⅛ inches, clothbound; available through MACHINE DESIGN, $4.50 *postpaid.*

For some obscure reason it has been only during the last decade that the true significance of industrial lighting has become generally recognized. Especially in this day when so great a percentage of the average worker's tasks are performed in an "arm's length" area, improper lighting tends to accelerate the trend toward more and more nearsightedness as well as other eye defects.

The machine designer will do well to explore the possibilities of reducing operator eye fatigue through the use of scientifically planned lighting, judicious use of color, and by designing work areas and machine controls so they can be quickly and clearly defined visually. It is an established fact that eye fatigue adversely affects not only the eyes but results in strain on the nervous system and overall mental and body fatigue. This whole subject of light, vision and seeing, with its many ramifications, is tied up so intimately with worker efficiency and health that its importance to all industries is indeed great.

Fully and clearly explained in this splendid book are the answers to just about every practical question concerning seeing in its relation to light and vision. The various phases of the subject are treated in a manner that is simple enough for direct application and thorough enough to give a broad scientific foundation. Complete data are presented in the simplest possible diagrams and discussions deal primarily with controllable aids to seeing. The author's writing style is excellent and makes for easy interesting reading.

Design Abstracts

Spark-Ignition Engines Compared

COMPARISONS of fuel-injection spark-ignition engines depend on whether the engines use oil or gasoline for fuel. Advantages claimed for spark-ignition fuel injection versus most carburetor manifold systems are:

1. Equality of fuel distribution directly to each cylinder over the complete load and speed range without hot spots or preheated air.
2. Excellent atomization of fuel, by use of from 200 to more than 1000 pounds per square inch at discharge nozzles instead of less than 14 pounds per square inch available to the carburetor.
3. Freedom from icing with elimination of local heat as at the idle system required by carburetors.
4. Ability to use a greater range of fuels in regard to volatility without special heat interchangers.
5. Greater safety since backfires are practically eliminated.
6. Two-cycle engines become possible with fuel economies approaching the 4-cycle engine.
7. Maximum power is increased, and fuel economy can be better. Engine becomes noticeably smoother, starting can be easier, and warmup quicker.
8. Since fuel is placed at the intake port or in the cylinder, acceleration is quicker and engine operation more flexible.
9. Exhaust odors can be eliminated when the engine is used as a brake by completely stopping fuel discharge.

There are some disadvantages. Cost is generally greater than for carburetor equipment. Reliability may not be considered as good as for the simpler carburetor, and maintenance expense may be greater due to more parts and greater sensitivity to dirt and corrosion. Vapor lock may be controlled, but with more difficulty than just venting of the float chamber of the carburetor.

If low volatility fuels are used, the engine must have provision to avoid cylinder-wall oil dilution with consequent excessive cylinder bore and piston ring wear. Provision must be made for a gasoline priming arrangement, or for switching to gasoline when stopping and starting.

Since good fuel distribution is inherent, manifold heat can be done away with. Temperature increase over atmospheric to help the carburetor induction system may be from 35 to 70 degrees Fahr. over the speed range at full load. Maximum power loss for this temperature increase in the high-speed range is generally not over 3 or 4 per cent. Maximum carburetor pressure drop at higher speeds usually does not exceed 2 inches of mercury. With fuel injection this pressure loss can be practically eliminated with a gain of about 6 per cent in maximum torque and power. Hence, a full power gain of about 10 per cent might be expected over a reasonably good carburetor manifolded type gasoline engine.—*From a paper by N. N. Tilley, Studebaker Corp., presented at the recent National War Materiel meeting of the S.A.E.*

Getting the Most Out of Materials

WHEN it is realized that typical engineering materials are contributing only a small fraction of their potential strengths, and when the reasons for such a state of affairs are understood, it may be possible to do something about it. Whereas tensile strengths should range in the millions of pounds per square inch for metals, ceramic materials, and resins, we encounter in practice only a few thousand pounds in most of these materials, or efficiencies of the order of .1 to .5 per cent. The theoretical tensile strength of phenol-formaldehyde resins is about 150,000 pounds per square inch. In practice, most resins of this type, fail at less than 3000 pounds per square inch.

Until quite recently most plastic laminates were produced from woven fabrics. The full strength of a tensile member is attainable only when it is perfectly straight with the applied load acting along its longitudinal axis. Some strength values reported by Dr. R. W. Webb of the U. S. Department of Agriculture, with cotton fiber bundles possessing different amounts of twist and tested by the Chandler bundle method, are of interest. Fiber bundles of Sea Island cotton were tested containing 1, 2, 3, and 4 turns per bundle, which calculates to be 1.6, 3.2, 5.33, and 8 twists per inch, respectively. The respective average fiber bundle strengths obtained were 93,500, 80,000, 65,900 and 48,000 pounds per square inch as compared with 98,800 pounds per square inch for the average of the control fiber bundles containing no twists. The figures cited show well how rapidly cotton fiber bundle strength is lost with insertion of increasing twist and with departure of the fibers from a straightened condition.

It is a fact that the day is gone beyond recall when we may do things by just simply hoping for the best. We must know exactly why we are doing them, as far as the utmost resources of science can tell us, so that we may control our manufacturing materials and not they us. Can we say that we have every process under strict control? We know too well that we cannot attain this, and perhaps it is too much to ask. But can we even say that we have our processes under such an amount of control as the available scientific knowledge permits? Again we confess that the answer is "no". There needs to be much closer cooperation between scientist and technologist than exists today.—*From a paper by Maj. Russell M. Houghton, Air Technical Service Command, presented at the recent National Aeronautic Meeting of the S.A.E. in Los Angeles.*

BUSINESS AND SALES BRIEFS

ORGANIZATION of a gasket and packing department in the industrial division has been announced by Armstrong Cork Co., Lancaster, Pa. In charge of the new department will be R. M. Hill who has been appointed assistant general sales manager. Assisting him will be Lewis Woobey as assistant sales manager and George Hodge as a specialist in synthetic rubbers.

♦

Sales representative for the past four years, John B. Girdler has been made sales manager for the Eastern district of Vanadium Corp. of America, and will make his headquarters at 420 Lexington avenue, New York.

♦

Succeeding the late Allen N. Bradford is Foster E. Fike as manager of the Rock Falls, Ill., plant of Russell, Burdsall & Ward Bolt and Nut Co., Port Chester, N. Y. Associated with the company for thirty-seven years, Mr. Fike's last position was that of sales manager.

♦

Moving of the general sales office of the Mechanical Specialties division from 500 Fifth avenue, New York 18, to Clinton, Mass., where the products of the division are manufactured, has been announced by Wickwire Spencer Steel Co. A district sales office of the division will be continued at the New York address.

♦

Clarostat Mfg. Co. Inc., Brooklyn, N. Y., has appointed Leon L. Adelman as advisory sales manager. Mr. Adelman's duties will also include acting as metropolitan New York sales representative.

♦

Opening of a new manufacturing plant and warehouse at 2310 Ranier avenue, Seattle 44, has been announced by Square D Co., Detroit. Walter H. Brodle has been named manager and will direct its operations.

♦

With offices at 412 Peoples Gas building, Pittsburgh, the D. D. Foster Co. has been appointed representative in this area for Hills-McCanna Co., Chicago.

♦

Formerly manager of sales activities in the Pacific region, Frank C. Angle has been named manager of all field sales offices of the General Machinery division of Allis-Chalmers Mfg. Co., Milwaukee. In his new position Mr. Angle will continue to supervise operations in the Pacific region.

♦

Transfer of B. P. Hess to the General Mill section of the Industry department at East Pittsburgh has been announced by Westinghouse Electric & Mfg. Co. Connected with rural ele-

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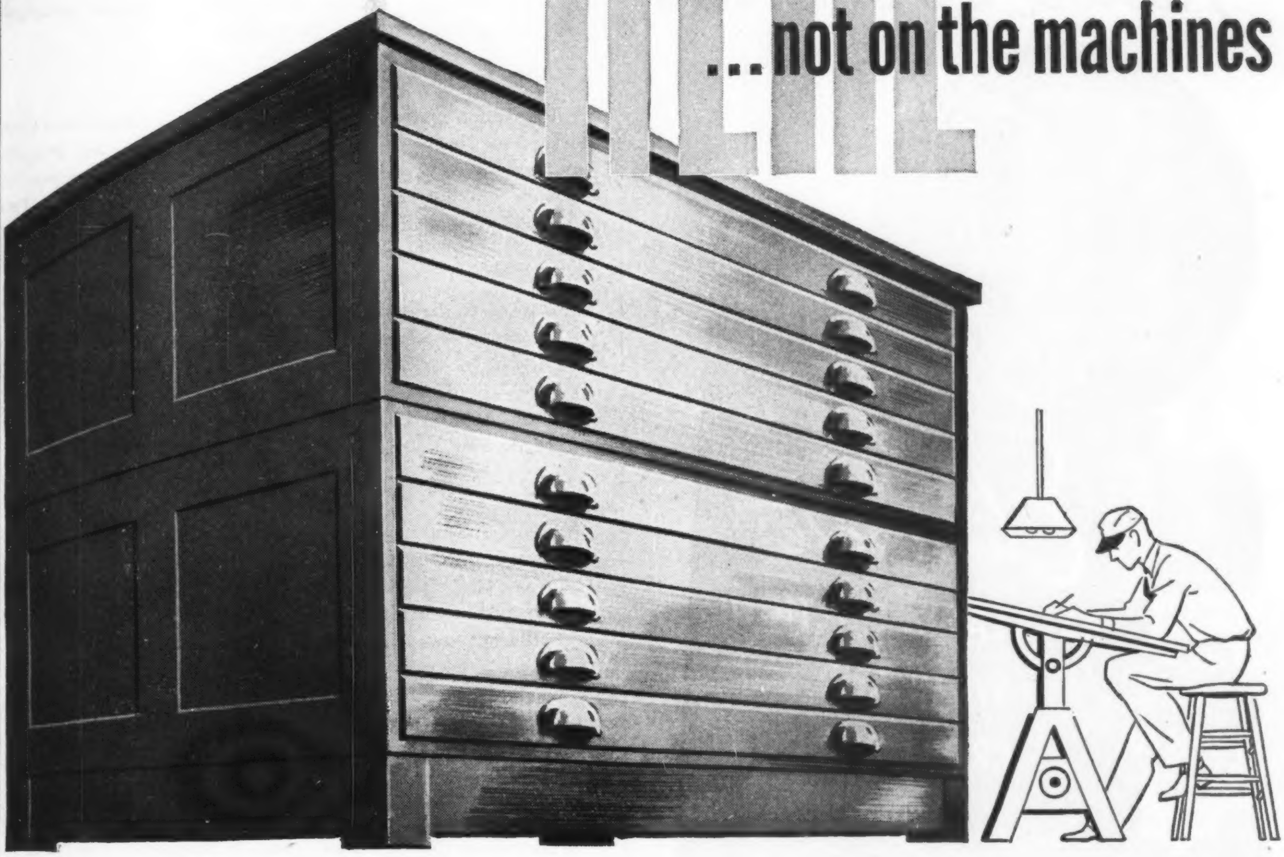

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trification activity in the Pittsburgh office for many years, Mr. Hess will inform farm equipment manufacturers of the latest developments in electric drive and control.

♦

According to a recent announcement, the Metal Fusion Corp. of America has been made a subsidiary of Cook Electric Co., Chicago. Position of manager of the new subsidiary has been given to Walter A. Ziebell.

♦

Acquisition of the properties of Empire Finished Steel Corp., located at Newark, N. J., and Putnam, Conn., has been announced by Wyckoff Steel Co., Pittsburgh. The property at Newark will be known as Empire Works while that at Putnam will be called New England Works.

♦

Howard C. Sauer has been named general manager of the new foreign division of The Timken Roller Bearing Co., Canton 6, O. In his new position Mr. Sauer will handle the sales and service of products—bearings, steel and detachable rock bits—in the world market outside the United States. Offices of the division will be in Canton, O.

♦

Opening of a new Kansas City branch at 421 Southwest boulevard, Kansas City 8, has been announced by General Controls Co., Glendale, Calif. With Robert Courtney in charge as branch manager, the new office will cover Kansas and adjacent areas in Missouri, Nebraska and Iowa.

♦

Littelfuse Inc. has appointed Allied Radio Corp., 833 West Jackson boulevard, Chicago 7, as a major distributor of its line.

♦

Succeeding Edward Stauverman is William J. Van Vleck as manager of the Atlanta office of Worthington Pump & Machinery Corp. Mr. Van Vleck had been assistant manager of the Philadelphia district office since 1938.

♦

Two field engineering representatives have been added to the Philadelphia branch personnel according to a recent announcement by Kennametal Inc., Latrobe, Pa. They are: Paul A. Herr, formerly a partner of Alfred Stauffer Machine Shops at Honey Brook, Pa.; and Harris H. Robbins, previously with St. Paul Hydraulic Hoist Co.

♦

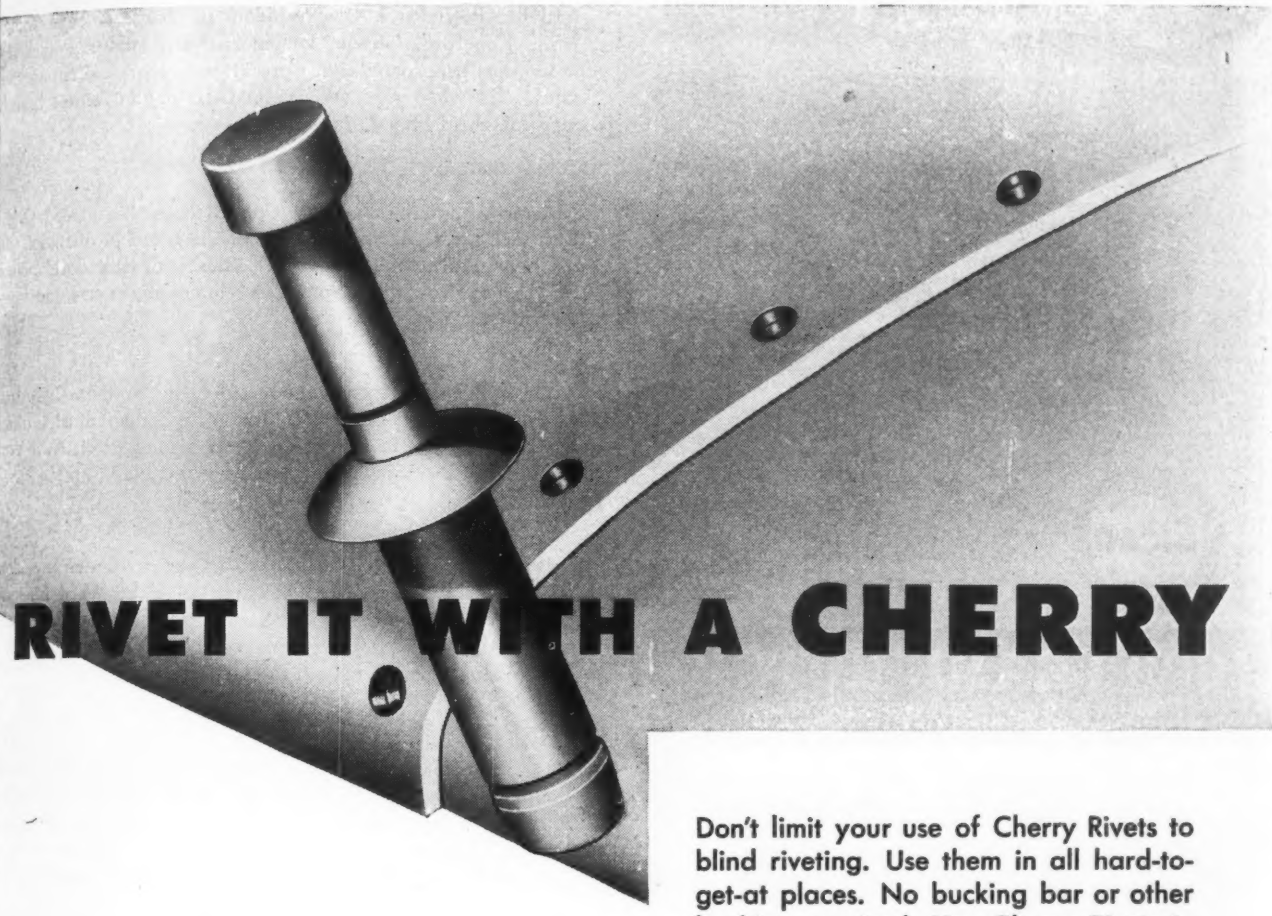
Plans for an addition to the production facilities at the Creighton, Pa., plant have been announced by Pittsburgh Plate Glass Co., Pittsburgh.

♦

According to an announcement by Latrobe Electric Steel Co., Latrobe, Pa., Victor F. J. Tlack has been appointed consultant and special representative of the sales department, and will make his headquarters at the Cleveland office located at 4516 Superior street. Prior to his appointment, Mr. Tlack had been president of Darwin & Milner Inc., Cleveland.

♦

Promotion of three executives to the position of vice president has been announced by The Weatherhead Co., Cleveland. Associated with the company since 1936, H. Church has become vice president in charge of sales. George H. Hof-



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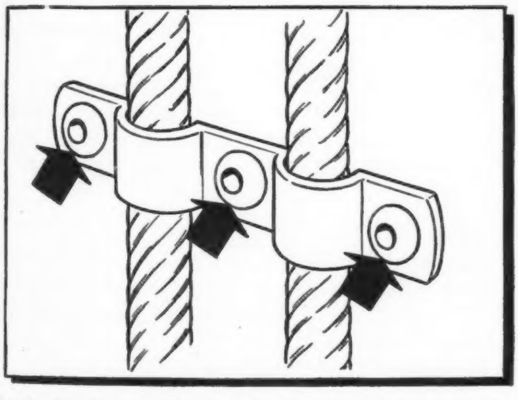

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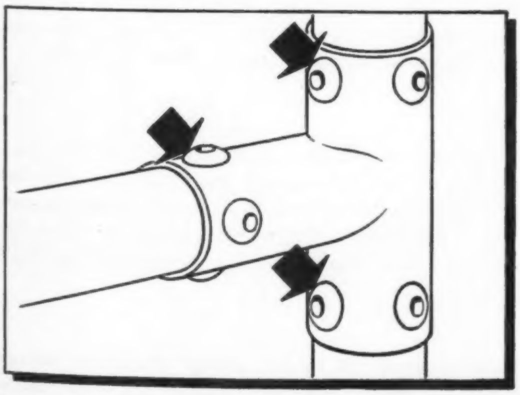

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LOS ANGELES, CALIFORNIA

ferd has been made vice president in charge of engineering while Robert P. Gibson, manager of the industrial sales division, has been named vice president in charge of automotive sales. He will have new headquarters at the Detroit branch office in the Fisher building.

♦

Acme Aluminum Foundry Co., Chicago, has named O. L. Earl vice president and member of the board of directors. In his new position he will direct sales and sales development for the company, which produces aluminum, bronze and magnesium castings.

♦

According to a recent announcement by Northern Industrial Chemical Co., South Boston, the metropolitan district office has been moved from 11 West Forty-second street, New York 18, to 1180 Raymond boulevard, Newark, N. J.

♦

Promotion of Arthur A. Berard to executive vice president and general manager has been announced by Ward Leonard Electric Co., Mount Vernon, N. Y. Having joined the company in 1920, Mr. Berard has served as salesman, general works manager, and general sales manager.

♦

Connected with Cutler-Hammer Inc., Milwaukee, since 1928, John Marvin Cook has been made manager of the San Francisco district sales office which covers northern California, western Nevada and part of Oregon. Mr. Cook has been in the Cincinnati district office for the past eleven years.

♦

Appointment of Littleton C. Barkley, former manager of the New York office, as sales manager of the Manhattan Mechanical Rubber Goods Sales department has been announced by The Manhattan Rubber Mfg. division of Raybestos-Manhattan Inc., Passaic, N. J. For the time being Mr. Barkley's office will be located at 120 Broadway, New York.

♦

Purchase of a 12-acre plot for a postwar manufacturing plant in Anaheim, Calif., is being planned by General Electric Co. The plant would be devoted to the manufacture of plastics parts for airplanes.

♦

Associated with the company for twenty-five years, Thomas R. Horan has been made sales manager of the Superseal Tube Fittings department of Grinnell Co., Providence, R. I. In his new position, Mr. Horan—who had been representative for the company in Washington for the past two years—will direct Superseal sales activities of fifteen branches.

♦

Opening of offices at Kansas City and St. Louis has been announced by Kennametal Inc., Latrobe, Pa. Both offices will be under the direction of R. B. Weeks, manager at Chicago. Ralph H. Craig has been placed in charge of the Kansas City office while Lyle H. Wade will head the St. Louis office.

♦

Advancement of Fred A. Koepf to district manager for the northwest Pacific division territory has been announced by Link-Belt Co., Chicago. Mr. Koepf will make his headquarters at Seattle. Succeeding him as assistant manager at

Los Angeles is George T. Lundquist who previously served as assistant to H. V. Eastling, vice president and sales manager, in San Francisco.

♦

Promotion of L. F. Weyand from general sales manager to general manager of the adhesive and coatings division has been announced by Minnesota Mining & Mfg. Co., St. Paul, Minn. In his new position Mr. Weyand will be responsible for all production, sales, research and laboratory activities of the division. He will continue to make his headquarters at the 3-M factory in Detroit.

MEETINGS AND EXPOSITIONS

Nov. 9-10—
Institute of the Aeronautical Sciences Inc. Fall meeting to be held in Dayton, O. Additional information may be obtained from headquarters of the Society at 30 Rockefeller Plaza, New York. Robert R. Dexter is secretary.

Nov. 9-10—
Society of Automotive Engineers Inc. National fuels and lubricants meeting to be held at Hotel Mayo, Tulsa, Okla. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

Nov. 13-14—
Society of the Plastics Industry. Annual fall convention and exhibit to be held at Waldorf-Astoria hotel, New York. Additional information may be obtained from headquarters of the Society at 295 Madison avenue, New York.

Nov. 15-18—
Society of Naval Architects and Marine Engineers. Annual meeting to be held at Waldorf-Astoria hotel, New York. J. H. King, 29 West Thirty-ninth street, New York, is secretary and treasurer.

Nov. 15-19—
American Chemical Society. Third national chemical exposition to be held at Chicago Coliseum, Chicago. M. H. Arveson, 330 South Wells street, Chicago, is chairman of the exposition committee.

Nov. 27-Dec. 1—
American Society of Mechanical Engineers. Annual meeting to be held at Hotel Pennsylvania, New York. Clarence E. Davies, 29 West Thirty-ninth street, New York, is secretary.

Nov. 27-Dec. 2—
Sixteenth National Exposition of Power and Mechanical Engineering to be held at Madison Square Garden, New York. Charles F. Roth, president of International Exposition Co., Grand Central Palace, New York, is manager of the exposition.

Dec. 11-13—
American Society of Agricultural Engineers. Fall meeting to be held at Hotel Stevens, Chicago. Raymond Olney, Box 229, Saint Joseph, Mich., is secretary.

Dec. 11-13—
American Society of Refrigerating Engineers. Fortieth annual meeting to be held at Hotel Pennsylvania, New York. David L. Fiske, 37 West Thirty-ninth street, New York, is secretary.

Jan. 8-12—
Society of Automotive Engineers Inc. Annual meeting and engineering display to be held at Book-Cadillac hotel, Detroit. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

Jan. 22-24—
American Society of Heating and Ventilating Engineers. Fifty-first annual meeting to be held at Hotel Statler, Boston. A. V. Hutchinson, 51 Madison avenue, New York, is secretary.